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DEVELOPMENT OF A SELF-CONTAINED
HEAT REJECTION MODULE (SHRM)

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PHASE I - FINAL REPORT

Report No. T211-RP-028

9 February 1976

Submitted by:

VOUGHT CORPORATION
Systems Division

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
HOUSTON, TEXAS

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1.0 INTRODUCTION AND SUMMARY

This is the final report submitted to NASA/JSC under contract NAS9-13533 in completion of the execution of that contract.

The effort involved in the Self-Contained Heat Rejection Module (SHRM) Phase I involved extensive conceptual design, analysis, design and test of a laboratory prototype and a preliminary design of a flight prototype SHRM. The analysis and design efforts were extensively documented in other reports which are referenced in Section 2.0 and will not be further documented in this final report. This report will discuss the laboratory prototype test hardware and testing.

The purpose and overall objective of the laboratory prototype test was to provide operational and design experience for application to a flight prototype design. The test conducted accomplished this and, in addition, provided test evaluation of several of the actual components which were to be used in the flight prototype hardware. Several changes were made in the flight prototype design due to these tests including simpler line routing, relocation of remote operated valves to a position upstream of the expansion valves, and shock mounting of the compressor. In addition, the Fairchild compressor was selected for use in the flight prototype test and its operational characteristics documented for use in test planning. The concept of heat rejection control by compressor speed reduction was verified and the liquid receiver, accumulator, remote control valves, oil separator and power source were demonstrated as acceptable for use. A problem was identified with the expansion valves and corrective action recommended. A procedure for mode changes between pumped fluid and vapor compression was developed and verified.

2.0 ANALYSIS, DESIGN AND COMPONENT EVALUATION

Extensive analyses were conducted under this contract and were thoroughly documented in the Appendices given in Section 5.0. In addition, element component development tests were conducted on the fluid swivel and contact heat exchanger. Table 1 lists the major analysis, design and component tasks conducted and the references in which the analyses are documented. The end item of the analyses and design efforts was the requirements and design of a laboratory prototype and flight prototype system.

TABLE 1

DESIGN, ANALYSIS AND COMPONENT EVALUATION TASKS

TASK	APPENDICES
Definition of Design Requirements and Groundrules	A
Thermal Environments Definition	B
Performance Parameters Calculations	C
Radiator Sizing and Performance	B,D
Refrigeration System Analysis	B,D
Structural Analysis	D
System Thermal Performance	D
System Weight Analysis	B
System Weight Optimization	B
Deployment Concepts Study	B
Fluid Swivel Design and Testing	B,F,G,I,J
Contact Heat Exchanger Design and Testing	B,C,E,F,H,I,J
Component Selection and Specification	D,F,J
Preliminary Design of SHRM Flight Prototype	A,C

3.0 TEST HARDWARE

The test hardware used for the laboratory prototype test was comprised of as many of the actual components being considered for use in the flight prototype program as were available at the time of the test. The remainder of the hardware was laboratory furnished equipment which was obtained or fabricated as needed for the test.

The objective of the hardware selection was to functionally simulate the flight prototype equipment as closely as possible. The refrigeration equipment used in this test was planned to be used in the flight prototype test also. Since development of specialized equipment was not possible commercially available components were purchased and used.

The schematics shown in Figures 1 and 2 and the photographs of Figures 3 and 4 illustrate the systems tested in the laboratory prototype test. The major flight prototype components to be evaluated were:

1. R-12 Compressors
2. Fluid Receiver
3. Liquid Accumulator
4. Expansion Valves
5. Remote Control Air Operated Valves
6. Oil Separator

Other major components used to simulate flight prototype equipment were:

7. Liquid Gear Pump to simulate flight centrifugal pump
8. Radiator Panel with auxiliary heat exchanger to simulate an array of four radiator panels
9. Tube-in-Tube Heat Exchanger to simulate the flight contact heat exchanger
10. Tube-in-Tube Heat Exchanger used for a precooler for the compressor

The equipment was arranged to operate either as a pumped fluid or a vapor compression system by selectively adjusting the valve positions and adding or removing refrigerant.

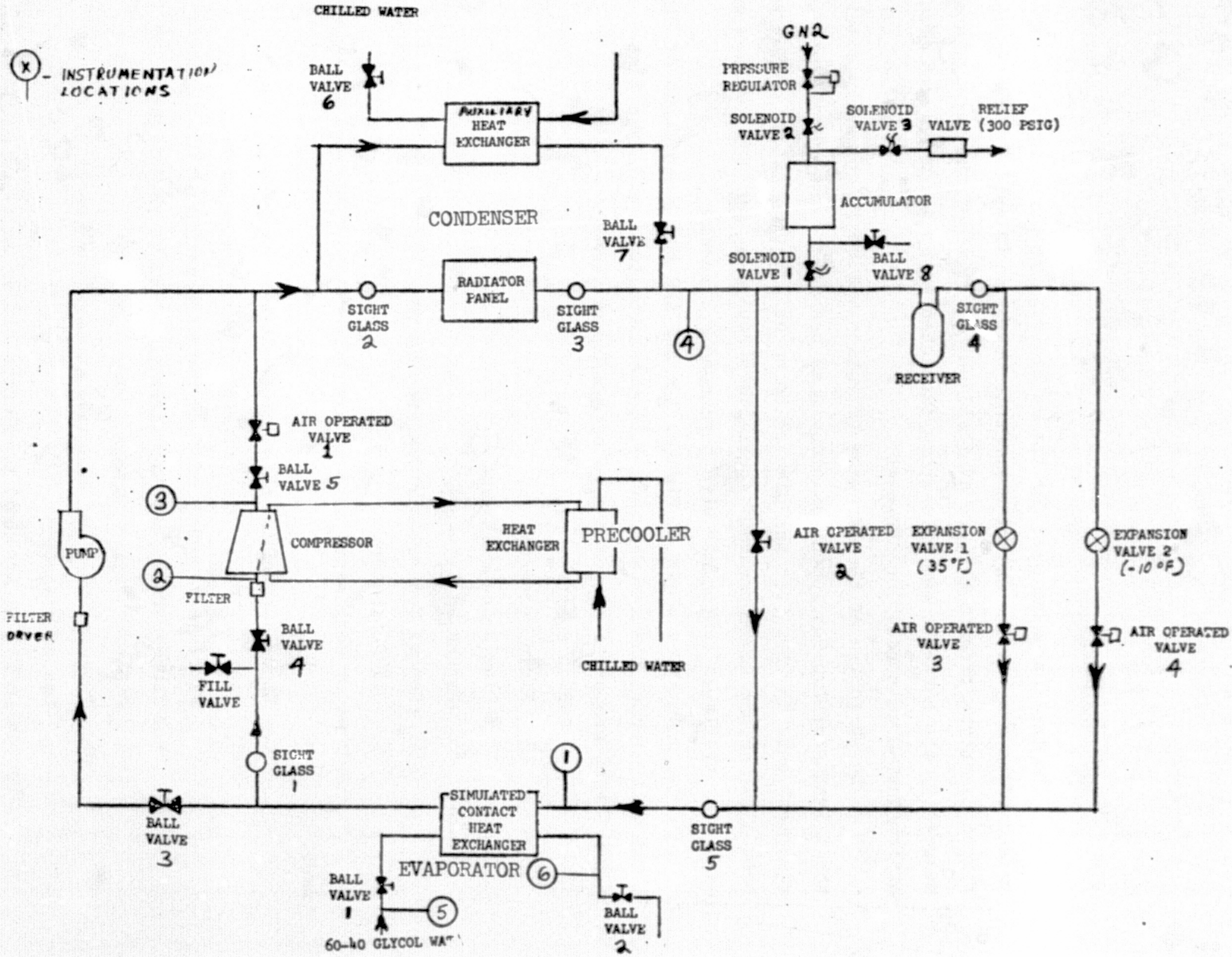

FIGURE 1 SHRM LABORATORY PROTOTYPE TEST SCHEMATIC WITH TASK COMPRESSOR

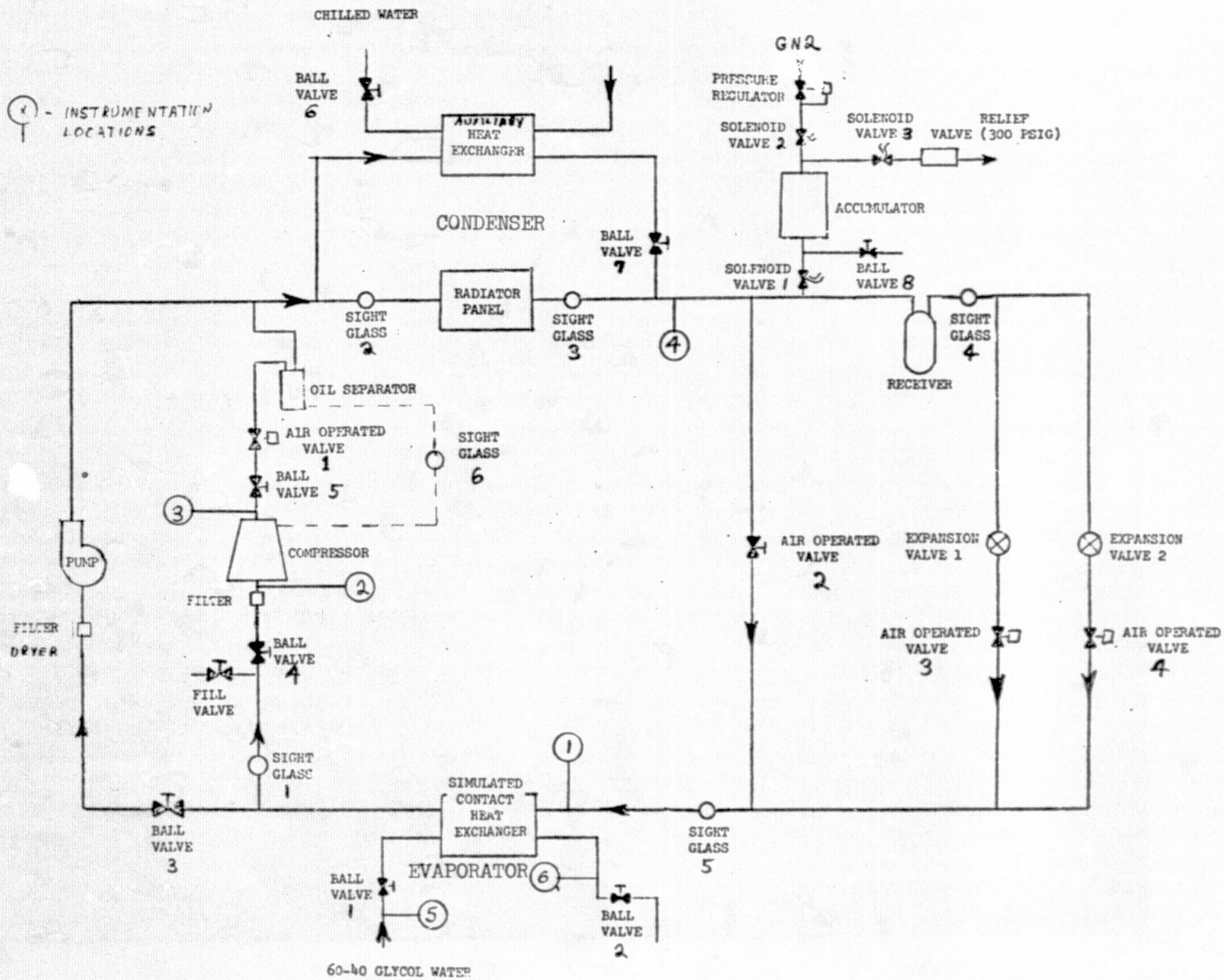

FIGURE 2 SHRM LABORATORY PROTOTYPE TEST SCHEMATIC WITH FAIRCHILD COMPRESSOR

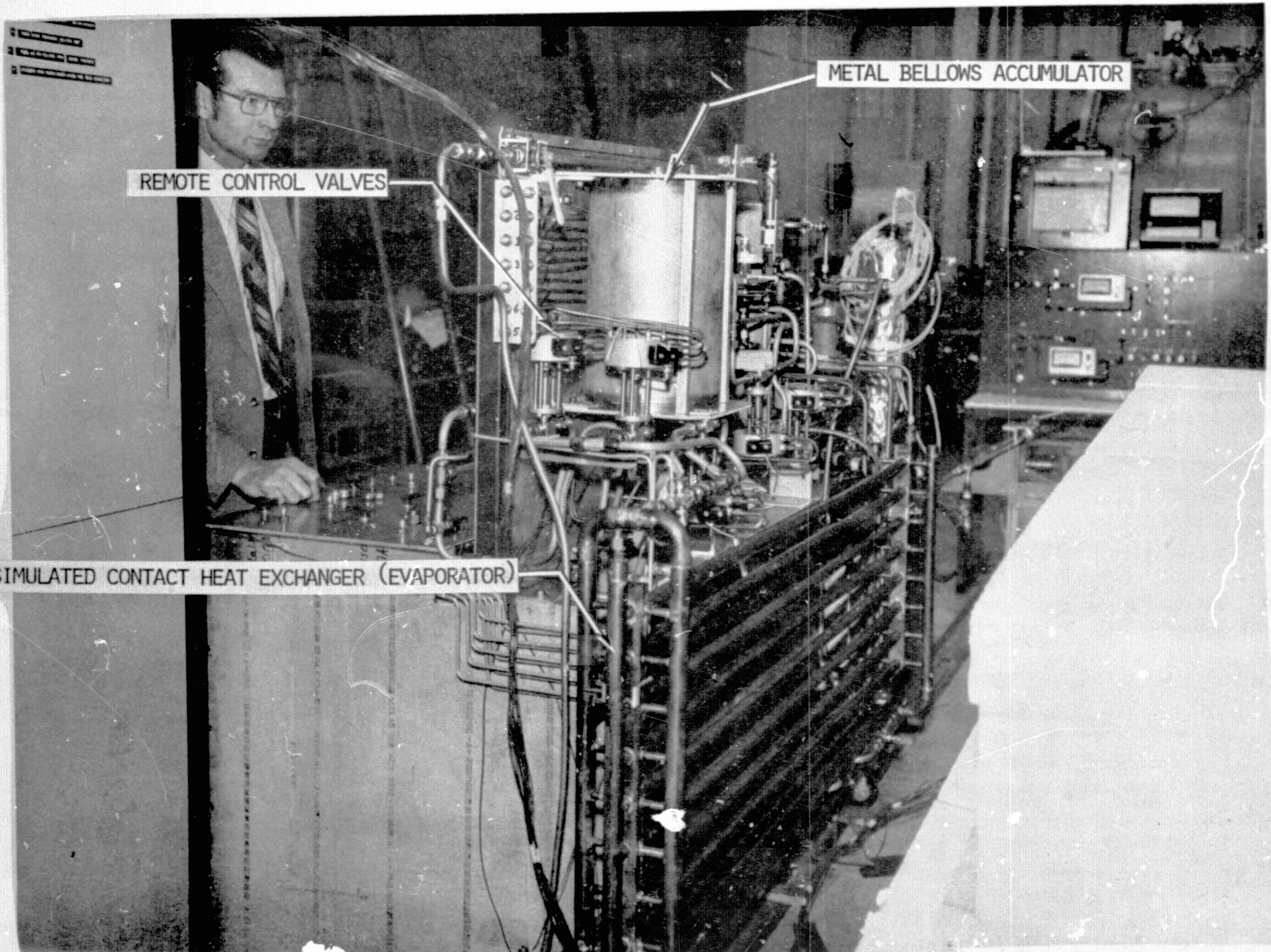

FIGURE 3 - LABORATORY PROTOTYPE TEST EQUIPMENT

Two R-12 compressors were evaluated in this testing; an aircraft rotary compressor manufactured by Task Corp. and an aircraft heli-rotor compressor made by Fairchild-Stratos. These two compressors are shown in the photograph of Figure 5. Both designs were commercially available but are low production quantity items and required a relatively long delivery time. The Figure 1 schematic shows the arrangement of the equipment for the Task compressor and Figure 2 for the Fairchild unit. As can be seen in these schematics there were some differences in test equipment necessary to accommodate the different compressors. The Task unit required a pre-cooler heat exchanger installed as shown in Figure 1; the purpose of which is to remove the discharge gas superheat. The internal arrangement of the Task compressor is such that the evaporator discharge passes into a motor inlet port where it passes over the motor for cooling purposes. The gas is therefore superheated to a greater extent than when it left the evaporator. The gas is then routed to the pre-cooler which is sized to remove all but 15°F or so of the superheat prior to returning to the compressor for the actual compression. For this demonstration test the necessary cooling in the pre-cooler was provided by a laboratory supplied chilled water supply. For a closed loop system it would be necessary to add this heat to the R-12 at some point in the loop. Probably the best location would be directly downstream of the evaporator and the system heat load and pre-cooler heat exchanger adjusted to return 15°F superheated vapor to the compression operation. The Fairchild compressor did not require a pre-cooler. The internal arrangement of this unit is a single pass which discharges superheated gas which includes motor waste heat and heat of compression. An oil separator is recommended for use with the Fairchild compressor to insure positive lubrication. An oil separator is not required for the Task compressor; this device has an internal oil separation chamber. Some hardware modifications were required prior to testing. The schematics of Figures 1 and 2 represent the final test configurations.

The fluid receiver is a commercially available, 12 lbm capacity tank manufactured by Standard Refrigeration. The device consists of a tank with an inlet port at the top and an outlet port connected to a standpipe which projects near the bottom of the tank. The purpose of the receiver is to insure the

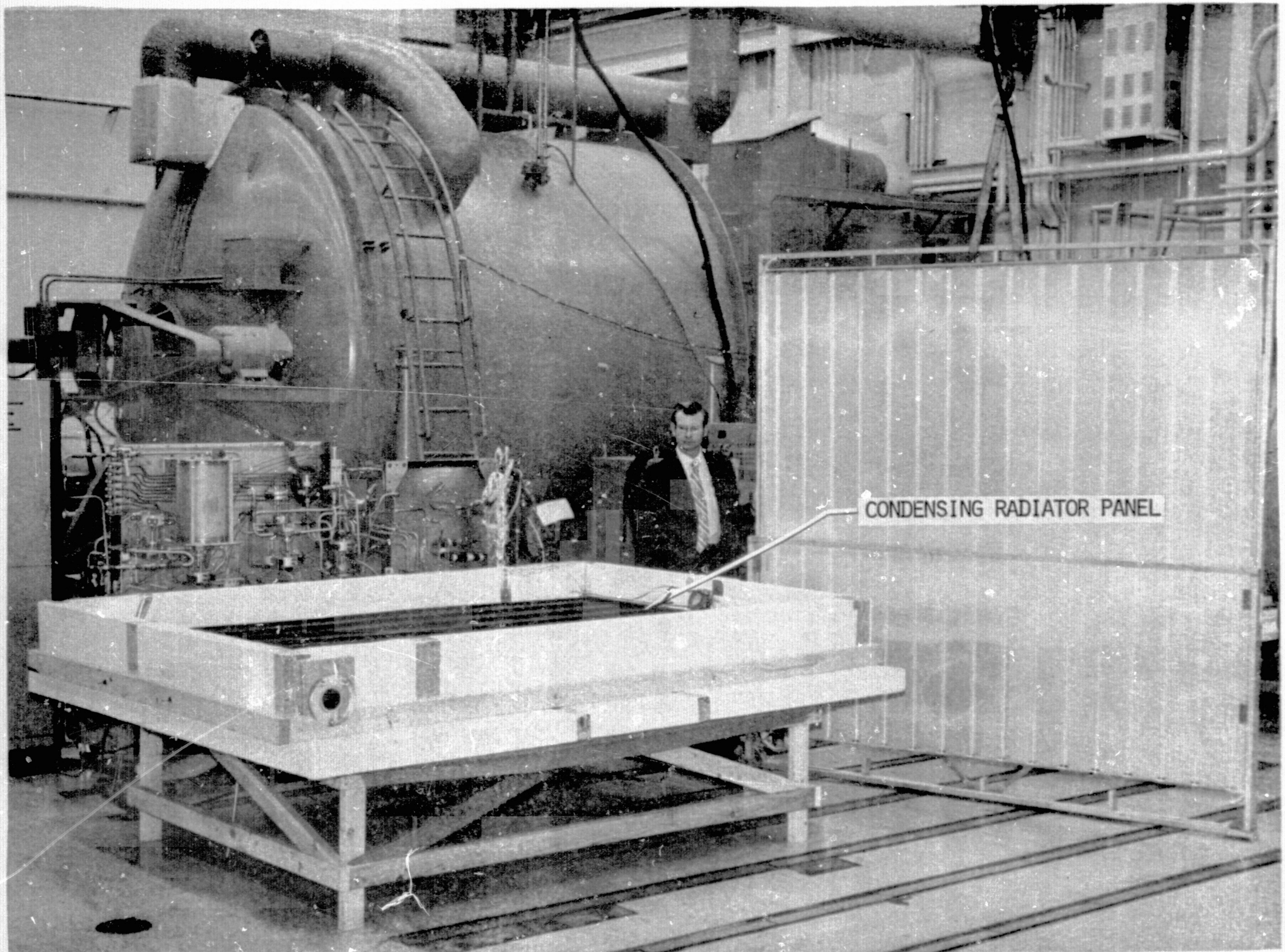

FIGURE 4 - LABORATORY PROTOTYPE TEST EQUIPMENT

expansion valve is in contact with liquid refrigerant at all times, even during transient conditions and over changes in average system temperature. In addition a receiver will permit the system to continue operation with small refrigerant leaks.

The liquid accumulator is a positive discharge metal bellows device manufactured by Metal Bellows Corp. This device consists of a cylindrical tank with ports on each end. A metal bellows provides 523 in^3 deliverable volume. The device also has a position indicator on the bellows which gives a fluid quantity measurement.

The two expansion valves are constant pressure valves manufactured by Refrigerating Specialties Co. These devices sense and control valve discharge pressure. Two valves are used to provide a selection between two evaporator pressures and therefore evaporator temperatures of 40°F and 0°F. Only the orifice sizes in the valves differ, one with a 1/8" orifice and one with a 3/16". The 1/8" orifice size was selected for the 0° evaporator temperature and the 3/16" for the 40° evaporator.

The remote control air operated valves are bellows type valves made by the Nupro Company. They are activated by pneumatic pressure which is controlled by three-way solenoid valves made by the Skinner Co. thus providing remote electrical control of valve position.

The oil separator used with the Fairchild compressor was a commercially available separator made by Standard Refrigeration. The device consists of a separation chamber into which the compressor discharge is routed at the top. The oil separates and collects in the bottom of the chamber and the vapor exits the separator at another port also at the top of the chamber. The oil that collects in the chamber is metered out of the separator by a float operated valve and routed to the compressor oil return port which is separated from the compressor suction plenum by an orifice. Oil flow is maintained by the compressor pressure rise.

The remainder of the hardware is conventional laboratory equipment and will not be further described with the exception of the radiator panel. The panel used in this test was a 6' x 8' panel constructed by seam welding 0.125 in. I.D. aluminum tube extrusions on an 0.063 in. aluminum sheet 8 inches apart. The extrusions were then manifolded together at each end of the panel. An auxiliary heat exchanger was connected in parallel with the panel to provide

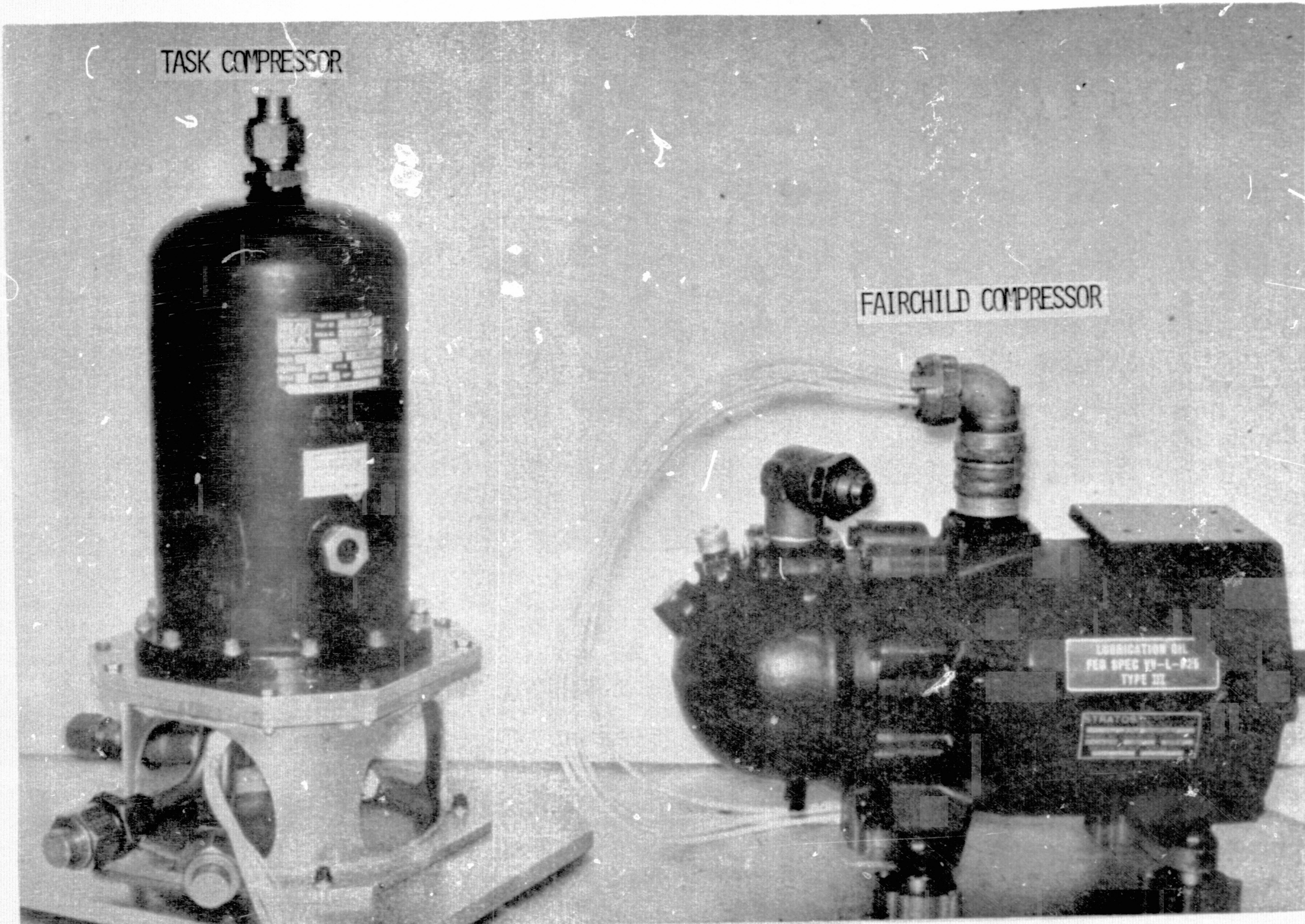

FIGURE 5 - CANDIDATE R12 COMPRESSORS

sufficient condensor heat rejection to test system capacity. Cooling of the auxiliary heat exchanger was provided by laboratory chilled water supply.

4.0 LABORATORY DEMONSTRATION AMBIENT TEST

The laboratory prototype test was unusually broad in scope. The general purpose of the test was to provide operational experience for designing, fabricating and operating the flight prototype SHRM. In addition, it was desired to evaluate and demonstrate as many of the system components to be used in the flight prototype as were available at the time of the test. The specific test objectives will be discussed in this section followed by a discussion of the test results and conclusions reached from the test results.

4.1 Test Objectives and Test Plan

The specific objectives of the laboratory prototype test were:

1. Demonstrate system operation in both pumped fluid and vapor compression operation.
2. Demonstrate refrigeration system operations at evaporator temperatures of +35°F and -10°F.
3. Generate performance data on the two compressors to allow selection of one for the flight prototype hardware and to assist in pre-test heat rejection predictions.
4. Demonstrate heat rejection control by compressor speed variation by input power frequency changes.
5. Evaluate the dual mode system mode changes.
6. Evaluate component operation of the:
   - (a) Fairchild and Task Compressors
   - (b) Fluid Receiver
   - (c) Liquid Accumulator
   - (d) Expansion Valves
   - (e) Remote control Air Operated Valves
   - (f) Oil Separator
   - (g) Variable Frequency/Variable Voltage Power Source
7. Develop procedures for servicing the system to be used with the flight prototype hardware.

The test plan to accomplish the above objectives was relatively simple and considerable lattitude was maintained in the procedure to allow system and procedure evaluation. The general scheme of the test plan was as follows:

1. Pre-Test Operations

    a. Fill system with liquid R-12 with accumulator approximately 1/2 full.

    b. Pressurize system to 150 psig by opening solenoid valve 1 and 2 and increasing GN_2 pressure on top of accumulator.

    c. Purge system using liquid pump circulating flow through filter dryer at pump inlet.

    d. Close SV1 to isolate accumulator and reduce charge to 30 lbm

    e. Initiate compressor operation and adjust to proper charge by monitoring sight glasses 3 and 4. When sight glass 4 is clear and 3 indicates a surface the system has proper charge. If sight glass 3 is clear remove R-12, if sight glass 4 is not clear add Freon. Monitor these throughout testing to insure proper charge.

2. Refrigeration System Testing

    a. Conduct the test points indicated on the test matrix given in Table 2. Pressure side pressure is to be adjusted by regulating the chilled water flow to the auxiliary heat exchanger to give 5 to 10°F subcooling at instrumentation location 4.

3. Mode Switch Testing

    a. Refill the system from the accumulator by opening solenoid valves 1 and 2 and increasing GN_2 pressure and run as pumped liquid. Investigate return to vapor compression mode operation by transferring liquid back to the accumulator.

In general this test plan was adhered to, however, considerable hardware problems were encountered in the operation of the system and the order of the testing was often modified.

TABLE 2 SHRM LABORATORY PROTOTYPE TEST MATRIX

TEST POINT	Glycol T_{in} (T20)	GLYCOL FLOWRATE (FM7) lbm/hr(%FM)	COMPRESSOR FREQUENCY (cps)	COMPRESSOR VOLTAGE (LEG-TO-LEG)	SUCTION SIDE PRESSURE (PSIG)	PRESSURE SIDE PRESSURE (PSIG)	EVAPORATOR TEMP. (°F)	CONDENSOR TEMP. (°F)
1	140	1000 (77)	400	208	32.6	157.65	35	120
2	100	1000 (77)	400	208	4.5	57.74	-10	60
3	140	1000 (77)	350	182	32.6	Adjust	35	Adjust
4	120	1000 (77)	300	156	32.6		35	
5	100	1000 (77)	250	130	32.6		35	
6	100	1000 (77)	200	104	32.6		35	
7	100	1000 (77)	150	78	32.6		35	
8	100	1000 (77)	108	52	32.6		35	
9	100	1000 (77)	350	182	4.5		-10	
10	100	1000 (77)	300	156	4.5		-10	
11	100	1000 (77)	250	130	4.5		-10	
12	100	1000 (77)	200	104	4.5		-10	
13	100	1000 (77)	150	78	4.5		-10	
14	100	1000 (77)	100	52	4.5		-10	

4.2 Test Results

Some of the more significant results of the laboratory prototype lay in the experience gained in the design and operation of a dual mode pumped liquid/refrigeration system. Continuous modifications to the system schematic were made beginning with checkout to improve the level of system performance. A system modification was made to lower pressure drop in the suction line when data indicated low suction pressure at the compressor inlet. Larger diameter lines (3/4 in.) were installed between the evaporator and the compressor and the line routing was simplified in order to achieve an acceptable pressure drop. The expansion valve orifice size was not large enough to achieve the 35°F evaporator and the two expansion valves were flowed in parallel to accommodate these test points. Larger orifice stems were available for these valves and were ordered for use in the flight prototype testing. The simulated contact heat exchanger used as the evaporator did not operate as efficiently as expected. As a result it was necessary to run higher heat source temperatures in the glycol/water system in order to add enough heat to the evaporator to define system capacity. The results and value of this experience are difficult to document but, in summary, this testing produced several changes in flight operation. In addition, procedures for servicing the system with refrigerant and compressor oil were developed and verified. The remainder of this section will discuss specific system test results in terms of performance with a recommendation for compressor selection for the flight prototype test.

An evaluation of the performance of the other components planned for use in the flight prototype will follow and the results of the mode switch experimentation will be discussed.

4.2.1 System Performance and Compressor Selection Recommendation

The system was operated at nominal evaporator temperatures of 35°F and 0°F over a range of compressor speeds with both compressors. The data from these tests are given in Tables 3 and 4 for the Fairchild and Task compressors respectively. The significant data from these tests with respect to compressor evaluation is the heat capacity at the two evaporator temperatures and the net heat rejection variation with compressor speed. A comparison of the heat rejection vs compressor speed curves is shown in

TABLE 3 SYSTEM PERFORMANCE WITH FAIRCHILD COMPRESSOR

INVERTER (RPM)	POWER FREQUENCY (CPS)	SUCTION PRESSURE (PSIG)	PRESSURE SIDE PRESSURE (PSIG)	EVAPORATOR TEMP. (°F)	CONDENSER TEMPERATURE (°F)	EVAPORATOR S.H. * (°F)	CONDENSER S.C. ** (°F)	Q_{REJ} (BTU/HR)
3400	400	33	157	36.2	120	11.6	9	28,600
3225	380	34	155	36.2	119	15.3	8	28,530
3000	355	34	155	35.0	119	16.0	6	25,800
2750	324	35	141	36.0	112	15.0	8	26,091
2500	294	34	129	38.0	106	12.0	6	24,332
2250	264	37	118	40.0	100.5	9.0	7.5	22,594
2000	234	38	103	42.0	92	9.0	7.0	22,451
1750	204	39	93	43	86	9.0	8.0	18,737
1500	174	42	90	45	82	11.0	11.0	16,033
1250	145	44	85	48	81	23.0	13.0	4,966
3400	400	10	92	5	85	18.0	13	10,086
3000	352	10	94	6	86	46.0	13	11,385
2750	324	11	92	7	85	19.0	13.0	10,278
2500	293	13	90	9	84	22.0	15.0	11,083
2250	264	14	89	12	84	13.0	19.0	7,950
2000	235	16	89	15	84	27.0	18.0	7,050
1750	204	18	85	17	81	23.0	12.0	7,040
1500	175	18	85	19	81	44.0	8.0	3,911

* S.H. - Superheat = Evaporator outlet temp (T2) - saturation temp at suction pressure

** S.C. - Subcooling = Saturation temp at pressure side pressure - condensor outlet temp (T4)

TABLE 3 (CONT'D) SYSTEM PERFORMANCE WITH FAIRCHILD COMPRESSOR

(Measurement Numbers Refer to Figure 2)

POWER FREQUENCY (CPS)	R-12 FLOWRATE (LBM/HR)	P1 (PSIG)	T1 (°F)	P2 (PSIG)	T2 (°F)	P3 (PSIG)	T3 (°F)	P4 (PSIG)	T4 (°F)	GLYCOL WATER FLOWRATE (LBM/HR)	T5 (°F)	T6 (°F)
400	545	33	36.2	25	47.8	157	250.4	149	111	1125	132.0	100.2
380	545	34	36.2	26	51.5	155	253.8	147	111	1125	132.2	100.5
355	535	34	35.0	25	51.0	155	243.	147	113	1075	136.0	106.0
324	520	35	36.	27	51.	141	212	131	104	1115	123.	93.
294	485	34	38.	30	50	129	208	122	100	1100	128	100
264	415	37	40	32	51	118	204	110	93	1100	127	101
234	365	38	42	37	51	103	176	100	85	1100	123	97
204	300	39	43	40	52	93	170	93	78	1085	122	100
174	225	42	45	42	56	90	156	88	71	1075	122	103
145	95	44	48	44	75	85	162	85	69	1075	99	93
402	255	10	5	5	23	92	270	92	72	1040	76	63
355	230	10	6	5	52	94	259	85	73	1070	90	76
324	215	11	7	7	26	92	259	85	72	1050	82	69
294	200	13	9	13	32	90	248	90	69	1050	84	70
264	175	14	12	14	25	89	239	90	65	1060	77	67
234	150	16	15	16	42	90	237	90	66	1050	74	65
204	115	17	18	50	18	85	230	85	69	1040	73	64
174	75	20	19	20	67	85	225	85	73	1050	73	68

TABLE 4 SYSTEM PERFORMANCE WITH TASK COMPRESSOR

INVERTER (RPM)	POWER FREQUENCY (CPS)	SUCTION PRESSURE (PSIG)	PRESSURE SIDE PRESSURE (PSIG)	EVAPORATOR TEMP. (°F)	CONDENSER TEMPERATURE (°F)	EVAPORATOR S.H.* (°F)	CONDENSER S.C.** (°F)	Q_{REJ} (BTU/HR)
3400	402	31.5	156	35	119	17	5	19,987
3000	355	32.5	141	37	112	14	3	18,876
2750	324	33.5	129	38	107	8	4	19,913
2500	294	37.0	135	38	110	13	5	18,298
2250	264	35.0	119	39	101	16	4	14,918
2000	234	36	111	40	88	9	0	14,436
1750	204	36.5	105	41	93	8	3	10,769
1500	174	38	95	42	87	10	1	11,578
1250	145	38.5	90	43	84	13	3	8,339
3400	400	7.5	86	-3	81	36	0	7,040
3000	352	7.5	88	4	83	27	3	4,023
2500	293	9.0	85	4	81	29	5	3,911
2000	235	10.0	82	7	79	29	5	6,831
1500	175	12	81	9	78	40	0	4,324

* S.H. - Superheat = Evaporator outlet temp (T2) - saturation temp at suction pressure

** S.C. - Subcooling = Saturation temp at pressure side pressure - condensor outlet temp (T4)

TABLE 4 (CONT'D) SYSTEM PERFORMANCE WITH TASK COMPRESSOR

(Measurement Numbers Refer to Figure 1)

FREQUENCY (CPS)	R-12 FLOWRATE (LBM/HR)	P1 (PSIG)	T1 (°F)	P2 (PSIG)	T2 (°F)	P3 (PSIG)	T3 (°F)	P4 (PSIG)	T4 (°F)	GLYCOL WATER FLOWRATE (LBM/HR)	T5 (°F)	T6 (°F)	PRECOOLER T_{in} (°F)	T_{out} (°F)
400	435	31.5	35.0	29.0	52.0	156	128	148	114	1100	123	100	180	120
355	405	32.5	37.0	30.5	51.0	141	121	140	109	1100	115	93	155	122
324	383	33.5	38.0	32.0	46.0	129	112	125	103	1100	112	89	134	113
294	383	37	38.0	32	51.0	135	123	130	105	1100	115	94	151	115
264	325	35	39.0	35	55.0	106	118	118	97	1100	104	86.5	115	108
234	300	36	40.0	35	49.0	111	102	110	93	1100	99	82		
204	260	36.5	41.0	36	53.0	105	97	105	90	1090	93	83	102	100
174	230	38.0	42	38	52	95	92	95	86	1090	95	81	96	94
145	200	39	43	39	55	90	87	90	81	1090	94	84	91	89
400	170	7.5	-3	5.0	33	86	153	86	81	525	78	60	207	87
352	170	7.5	4	6.0	33	88	140	88	80	540	77	67	192	88
293	165	9.0	4	7.0	33	85	121	86	76	525	77	67	164	84
235	150	10.0	7	7.5	36	82	97	82	24	655	75	61	131	82
175	125	12.0	9	10.0	49	81	86	81	80	645	75	64	109	83

Figures 6 and 7. As can be seen from these comparisons, the Fairchild compressor provides a significant amount more heat rejection than the Task compressor. At the rated power frequency of 400 Hz the Fairchild unit delivered 28,600 BTU/hr cooling in the evaporator, compared to 20,000 BTU/hr for the Task unit. Nominal performance curves provided by the manufacturers indicated heat rejection of 39,500 BTU/hr for the Fairchild and 30,000 BTU/hr for the Task compressor could be expected at a 120°F condenser and a 35°F evaporator temperature. Neither unit performed up to these expectations.

The Fairchild compressor delivered 72% of the expected performance, and the Task compressor delivered 66%. Vendor supplied data had shown an efficiency advantage for the Task unit which was indicated to deliver 7690 BTU of cooling per kilowatt of input power and the Fairchild 6800. Power input to the compressors was not measured during these tests but if each had shown the expected power at 120°F condenser, 35°F evaporator the cooling per kilowatt would have been about 5000 BTU per kw for both compressors. At the 35° evaporator temperature both compressors indicated heat rejection control was possible by reducing compressor speed through a power frequency reduction. Better than 2 to 1 heat capacity range was observed with both compressors. An intense high frequency vibration was observed when operating the Task compressor. Vendor representatives indicated this was a characteristic of the device. The vibration necessitated shock mounting of the compressor to protect other components attached to the flow bench and use of flex lines for flow connections to prevent work hardening and subsequent fatigue failure of lines. This type of vibration was not present with the Fairchild compressor, however it was recommended from these tests to shock mount the compressor for the flight prototype system.

There was no performance data available for a 0°F evaporator temperature from either compressor manufacturer. Data taken during these tests are also shown in Figures 6 and 7. As seen the Fairchild compressor indicated about 4000 BTU/hr higher capacity than the Task unit. In addition, the Task performance was somewhat erratic as the speed was reduced having a range of 1.75:1 where the Fairchild had a 2.2:1 range with somewhat more predictable heat rejection characteristics.

Both of the compressors were tested with identical equipment with the exception of the precooler and oil separator discussed previously.

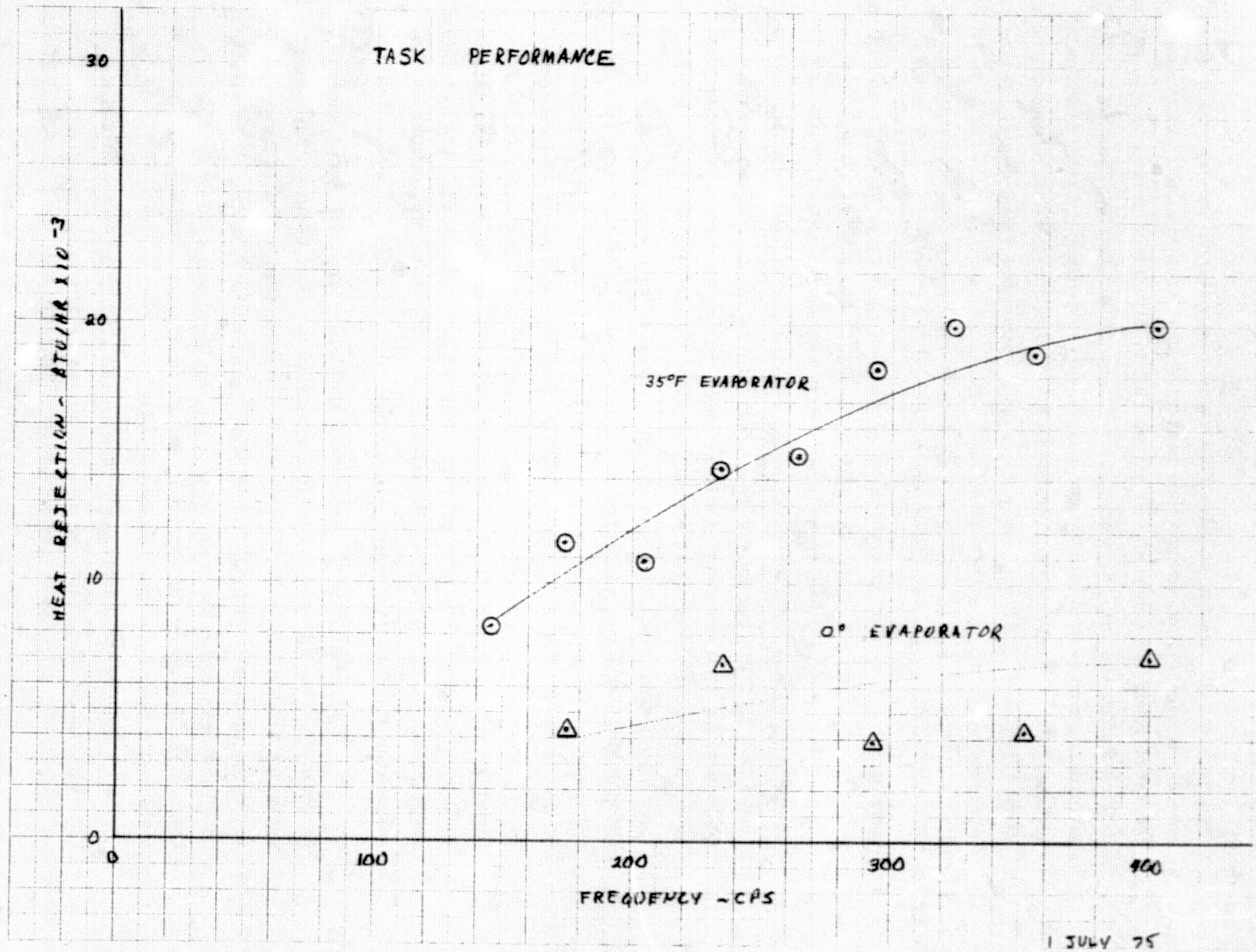

Figure 6 System Heat Rejection of Task Compressor

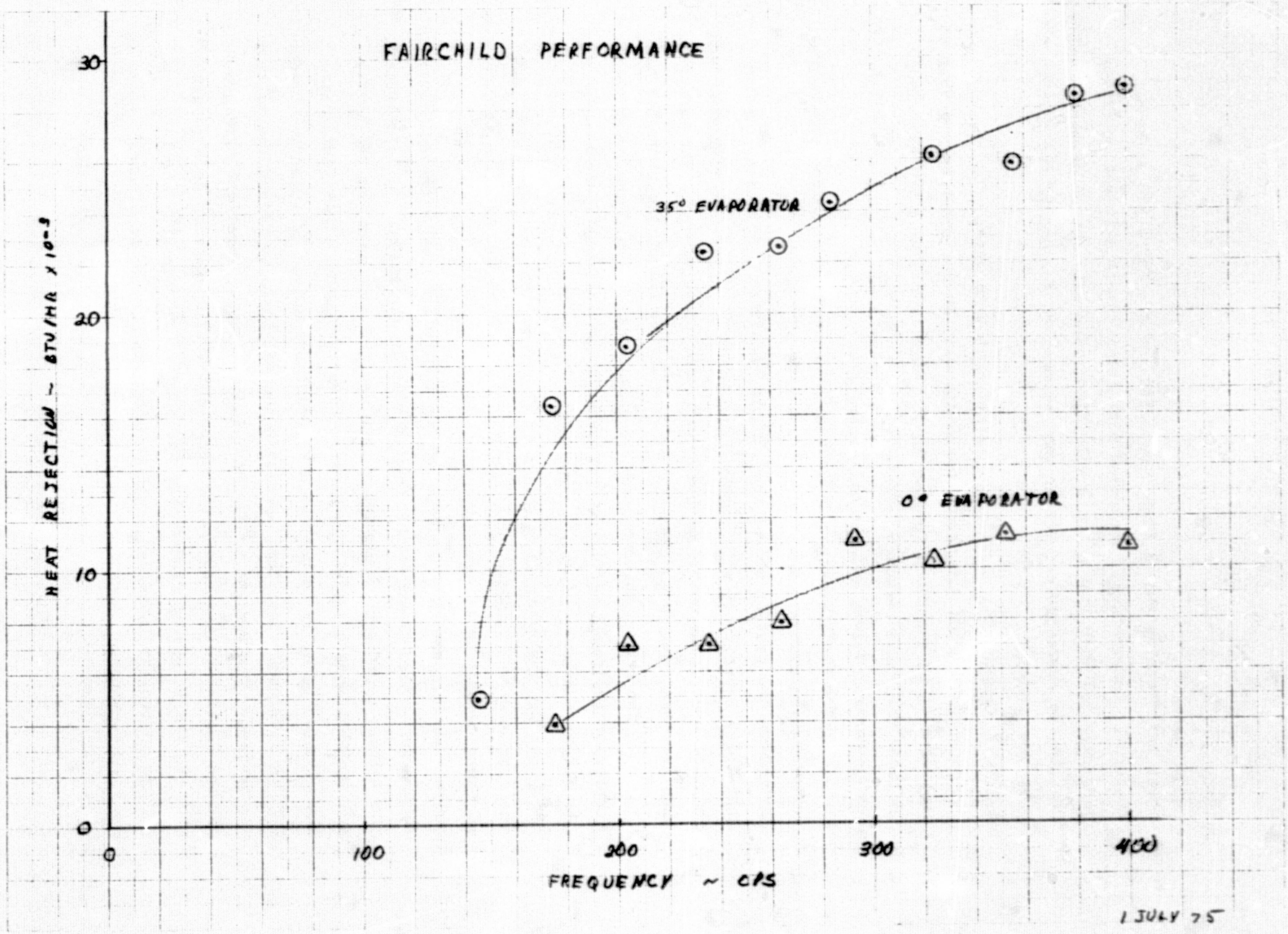

Figure 7 System Heat Rejection of Fairchild Compressor

These test results show a higher capacity for the Fairchild at both evaporator temperatures, increased simplicity due to elimination of the precooler and much less intense vibration. Therefore,based on these test results,the Fairchild compressor was recommended for use in the flight prototype program.

4.2.2 Component Evaluation of Flight Prototype Components

Several of the other components used in the laboratory prototype test were to be evaluated for use in the flight prototype system. These included the fluid receiver, rotary inverter, accumulator, expansion valves, remote control valves and the oil separator.

The fluid receiver indicated sufficient capacity for use in the refrigeration system. Monitoring sight glasses 3 and 4 (see Figures 1 and 2) throughout the testing showed that if the receiver is filled to a point just below capacity it will supply liquid to the expansion valve over the range of heat loads tested at both evaporator temperatures. The receiver, thus serviced, should be adequate for use in the flight prototype hardware.

The rotary inverter used to provide the three phase power to the compressor indicated sufficient performance and reliability to be used in the flight prototype hardware. The major function which was evaluated in this test was the controller system which reduced the voltage linearly as the frequency was reduced. Two controllers were evaluated. The performance of these devices is illustrated in Figures 8 and 9. Figure 8 shows the voltage variation with frequency for the controller used with the Fairchild compressor. Also shown is the desired voltage variation. As illustrated, the voltage decreased more quickly than desired when measured leg to leg and somewhat more quickly when measured leg to common. This controller was adjusted manually to achieve a constant current as frequency was reduced. For the Task compressor test automatic control logic to reduce the voltage was devised. The results in Figure 9 indicate quite excellent agreement between desired voltage characteristics and that measured. As a result of this testing it was concluded that the rotary inverter with the latter voltage controller installed would be adequate for use as a power source in the flight prototype test.

FAIRCHILD COMPRESSOR

3Φ POWER

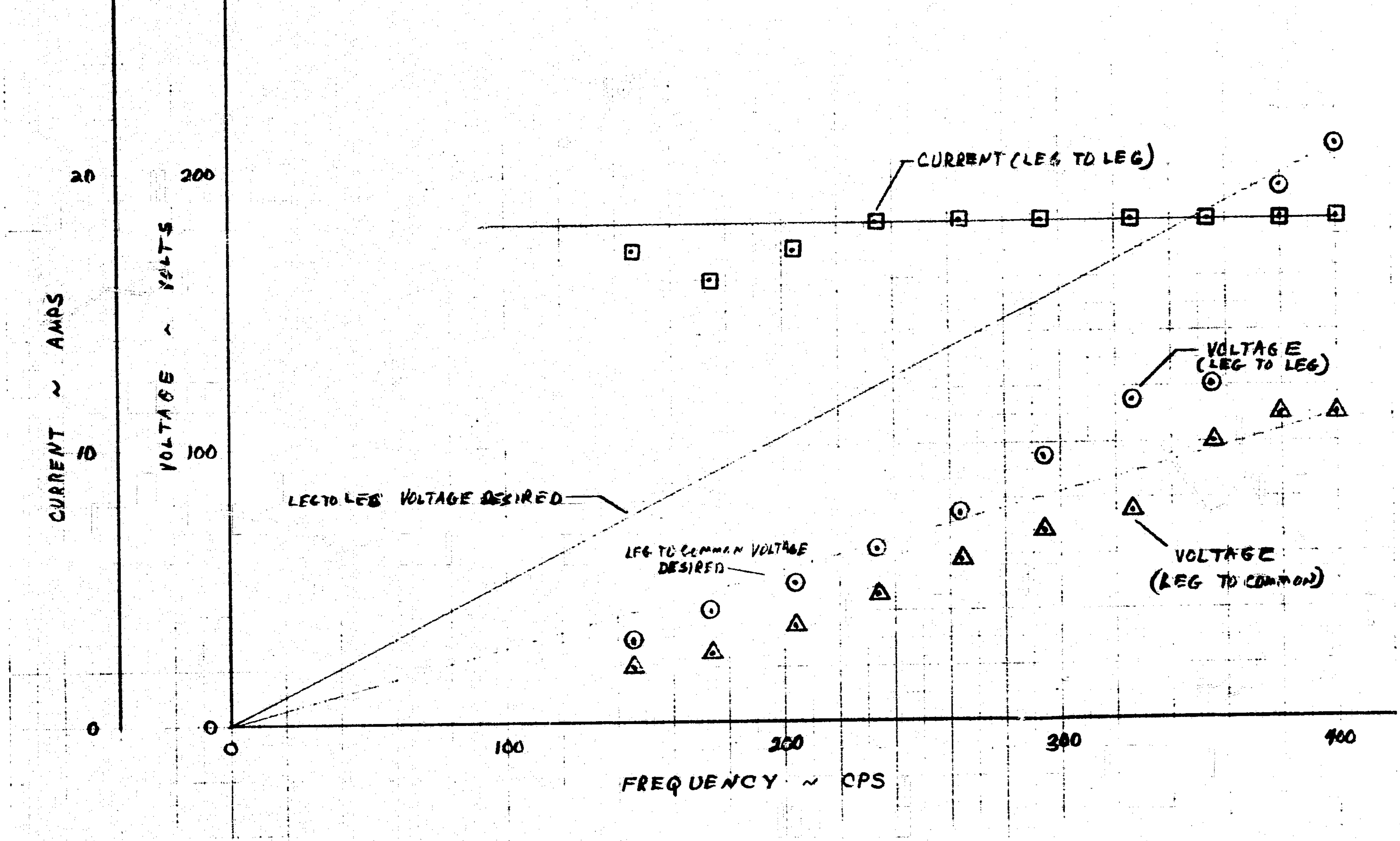

FIGURE 8 – VOLTAGE CONTROLLER PERFORMANCE - MANUAL   15 JUNE 1975

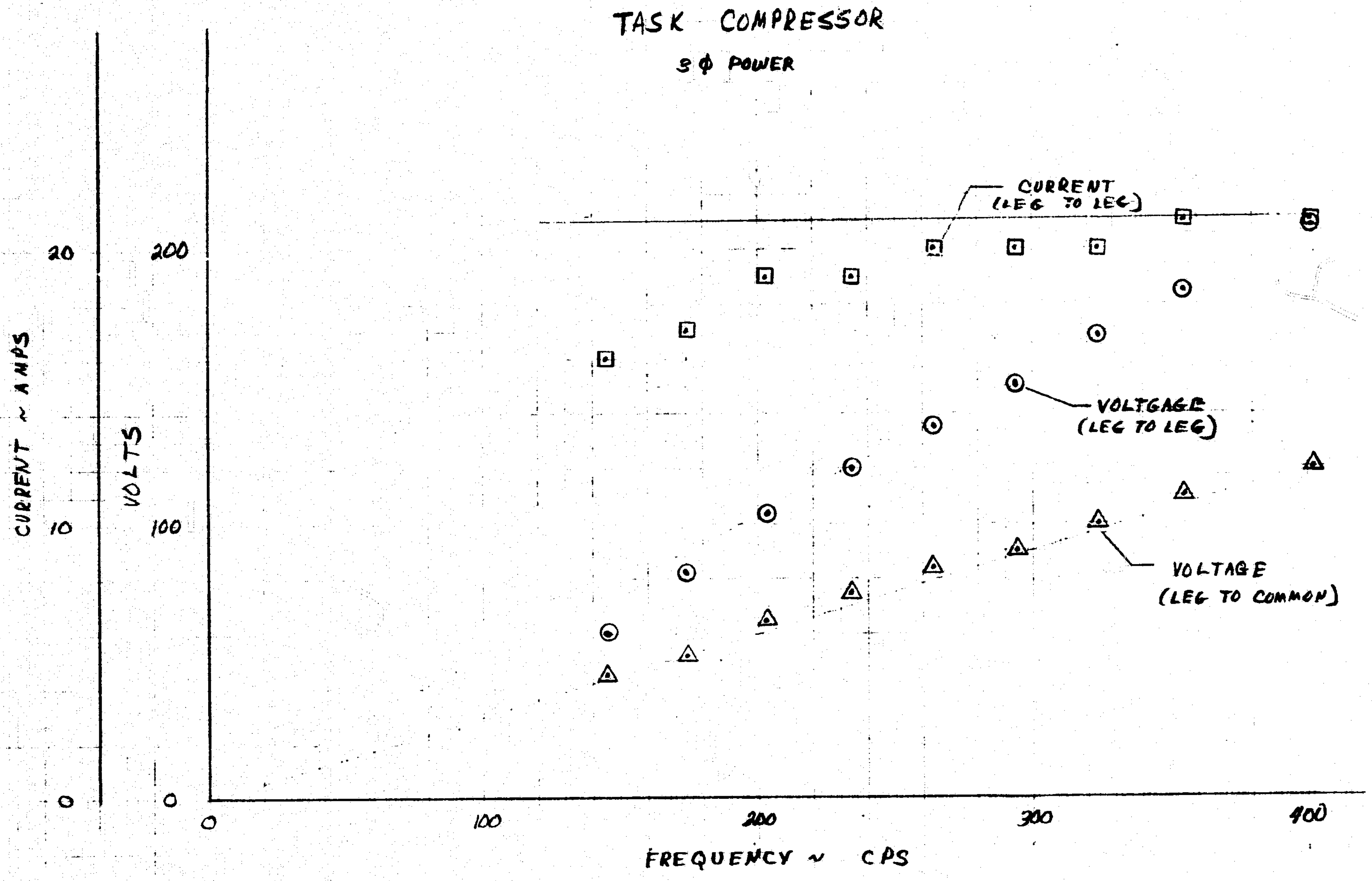

FIGURE 9 - VOLTAGE CONTROLLER PERFORMANCE - AUTOMATIC 19 JUNE 1975

The metal bellows accumulator, when first installed in the flow bench, indicated two defects; a leak from the top fitting seal and a failure to read out position indication. The accumulator was disassembled and repaired successfully and operated nominally for the remainder of the testing. This component was deemed functionally satisfactory for use in the flight prototype hardware.

The constant pressure expansion valves were operational functionally. The valve with the 1/8" orifice was used for the low evaporator (0°F) case and regulated evaporator temperature within 10°F over the operational range of the compressor. The characteristic of the operation was for the evaporator temperature to rise as compressor speed was reduced. This characteristic results from decreased pressure drop through the expansion valve as flowrate is reduced. The valve with the 3/16" orifice could not be set for the 35°F (33 psig) evaporator due to excessive pressure drop through the valve. For the purpose of this test the two valves were operated in parallel and adjusted to 33 psig. This arrangement proved successful; however, the increase in evaporator temperature as compressor speed was reduced occurred at this evaporator temperature also. There are larger size orifices available for installation into these expansion valves. It was recommended as a result of this test that the larger orifices be acquired and installed in the expansion valve to be used for the 35°F evaporator prior to installation into the flight prototype system.

Seven remote control valves were installed in the system. Four were bellows type air operated valves which were planned for use in the flight prototype system. The operation of these valves was evaluated during these tests and were found to have satisfactory response and reliability for use in the flight prototype system.

During the testing of the Fairchild compressor the oil separator was evaluated and found to provide positive oil return to the compressor at all the operating conditions tested. The device was judged to be satisfactory for use in the flight prototype system.

4.2.3 Mode Switch Experiments

The number and extent of the mode switching experimentation was curtailed somewhat by various malfunctions and system reworks; however, enough mode switching was accomplished to determine the adequacy of the technique of adding and removing R-12 by use of the metal bellows accumulator. It was found that flooding the system with R-12 to switch from vapor compression to pumped liquid operation by pressurizing the top of the accumulator from a regulated GN_2 worked quite satisfactorily. Removing the R-12 from the system proved to be more difficult. Simply venting the GN_2 from the top of the accumulator resulted in little transfer of liquid back to the accumulator. The bellows expanded but due to gaseous rather than liquid R-12. Heating the evaporator with the heat load source assisted some but once the liquid was driven from the evaporator the liquid transfer again stopped. In order to transfer the liquid back to the accumulator it was necessary to start the compressor with the valve to the accumulator open. This, fortunately, was successful since neither the rotary or heli rotor compressors are succeptible to damage by liquid slugging. Starting the compressor with excess R-12 in the system caused some start up power drains with high current, however, these were not damaging to the equipment and can be accommodated by the power source. The technique for mode switches appeared to be to simply open the valve to the accumulator and pressurize to fill the system with both the pump and compressor off. When the pressures around the system equalize with the accumulator pressure the system is full and the pump can be started for pumped liquid operation. To remove the liquid from the system for vapor compression operation the pump should be turned off, the accumulator vented to atmospheric pressure and a delay taken to allow as much liquid to transfer back into the accumulator as possible. The compressor should then be started and the accumulator pressure adjusted to a pressure higher than the vapor pressure but lower than the pressure side pressure to provide a ΔP to force the liquid from the condenser into the accumulator. When the level indicator indicates the proper amount has been removed from the system close the valve to the accumulator to isolate it from the system for vapor compression operation.

4.3 Conclusions

From the results of the testing both of the compressors indicated lower than expected performance. The Fairchild compressor had overall better performance and was recommended for use in the flight prototype system. The fluid receiver, rotary inverter, liquid accumulator, remote control valves, and oil separator performed satisfactorily in their as tested configurations and were judged to be acceptable for the flight prototype test. The low evaporator temperature (low pressure), 0°F , expansion valve operated satisfactorily, however, the higher temperature (higher pressure), 35°F valve failed to have sufficient adjustment range to regulate at the desired temperature. A larger size orifice has been ordered for installation into this valve prior to use in the laboratory prototype test. A method to switch modes by moving liquid R-12 into and out of the system using the accumulator and compressor was developed and experimentally verified. Considerable experience in refrigeration system design and operation for this unusual application was achieved and procedures for servicing the system with R-12 and compressor oil were developed and verified.

5.0 APPENDICES

A. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 21 June 1973 to 28 August 1973, Report No. T211-RP-001, 28 August 1973.

B. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 29 August 1973 through 28 September 1973, Report No. T211-RP-002, 2 October 1973.

C. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 29 September through 31 October 1973, Report No. T211-RP-003, 2 November 1973.

D. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 1-30 November 1973, Report No. T211-RP-004, 7 December 1973.

E. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 1 December 1973 through 21 January 1974, Report No. T211-RP-005, 22 January 1974.

F. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 22 January through 7 March 1974, Report No. T211-RP-007, 8 March 1974

G. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 8 March through 8 July 1974, Report No. T211-RP-008, 10 July 1974.

H. Joint Conductance Element Tests for the Self Contained Heat Rejection Module Contact Heat Exchanger, Report No. T211-RP-009, 15 August 1974.

I. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 9 July to 27 September 1974, Report No. T211-RP-010, 27 September 1974.

J. Development of a Self Contained Heat Rejection Module, Progress Report for the period of 27 September 1974 through 6 January 1975, Report No. T211-RP-012, 10 January 1975.

APPENDIX A

DEVELOPMENT OF A SELF-CONTAINED
HEAT REJECTION MODULE

Contract No. NAS9-13533
Report No. T211-RP-001

PROGRESS REPORT
FOR THE PERIOD OF 21 JUNE 1973 TO 28 AUGUST 1973

28 August 1973

Submitted by:

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORP.
DALLAS, TEXAS

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
Houston, Texas

Prepared by:

J. L. Williams

J. L. Williams

Approved by:

R. J. French

R. J. French, Supervisor
EC/LS Group

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1.0 INTRODUCTION

This first progress report on Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 21 June 1973 through 28 August 1973.

Effort during this period was primarily concerned with initiation of the study and establishment of a set of design requirements and a milestone schedule consistent with NASA-JSC's plans.

2.0 DISCUSSION OF WORK PERFORMED DURING THIS REPORTING PERIOD

2.1 Kick-Off Meeting

The kick-off meeting for the program was held at NASA-JSC on 1 August 1973. Attendees at the meeting were:

Wil Ellis	NASA-JSC
Burrell French	NASA-JSC
Dave Baskett	VSD
J. L. Williams	VSD

Discussion in the meeting indicated that: (a) The primary emphasis during the study will be on development of a system which is self-contained, and which is modular. The objectives are to reduce vehicle integration costs for the heat rejection system, and to reduce hardware costs through commonality which will allow larger production runs, and re-use of equipment. (b) During the analysis portion of the effort, a wide range of concepts (beyond that given in Reference [1]) will be considered and documented. This will include concepts for contact heat exchangers, dual mode refrigeration systems, combination pump/compression, etc. Factors such as technical risk, cost and schedule related to each component concept will be considered at the time of selection of concepts to be fabricated. Attractive concepts which are not selected due to these factors will also be identified. For example, the contact heat exchanger, which is very desirable because it obviously reduces vehicle integration problems, may require development beyond the scope of this study. If that is the case, then a conventional heat exchanger will be used in the test hardware, and the contact heat exchanger will be left for future development. (c) VSD should define more explicitly the analyses to be performed under the contract study. (d) A milestone chart schedule be established which shows when specific interface documents are to be transmitted to NASA-JSC. Generally, analyses results will be included in monthly progress reports. (e) The Self-Contained Heat Rejection Module (SHRM) should be designed to operate and to allow feasibility to be demonstrated in a one-gravity field as much as possible. The ground test conditions will be considered in structural design of deployment mechanisms, for example. (f) The design requirements should be rewritten to reflect applicability to a wide range of missions with various

temperature levels and heat rejection demands. In general, the shuttle orbiter should be used as a baseline for launch environments, and external environments during flight should include earth orbital, lunar orbital, translunar, and deep space conditions. Silver-backed Teflon will be considered for the radiator coating during sizing analyses. (g) The SHRM test article may use Refrigerant 12, or any other convenient fluid which is representative of Refrigerant 12. If a fluid other than Refrigerant 12 is selected for the flight SHRM, then justification should be provided.

2.2 Design Requirements and Ground Rules

The design requirements and ground rules as revised based on the kick-off meeting are given in Table I. This information is submitted for NASA-JSC concurrence and approval.

2.3 Schedule

The program schedule has been revised as shown on Figure 1. Program milestones are included on the schedule.

2.4 Planned Analyses

The analyses planned for the concept evaluation and design phase are:

(a) Thermal Environments - Establish the equivalent incident heat flux for each candidate mission. This will include consideration of solar radiation, planetary albedo and emission, and energy from adjacent structures where appropriate.

(b) Establish performance parameters, including maximum allowable pressure drop, power penalty, water availability, and component weight goals.

(c) Interface Heat Exchanger Sizing and Performance - Determine the size and performance required for each mechanical heat exchanger. Also perform sizing analysis on a fluid-to-fluid heat exchanger for backup and comparison.

(d) Radiator Sizing and Performance - Size radiators for each mission. Consider one and two-phase flow radiators. Also consider one and two-dimensional panel configurations.

(e) Refrigeration System Analyses - Perform sizing and performance analyses on refrigeration system. Consider the use of sophisticated expansion devices to conserve energy.

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TABLE I

SHRM DESIGN REQUIREMENTS AND GROUNDRULES

VEHICLE INTERFACE

	Low Return Temp.	Moderate Return Temp	High Return Temp
Return Temperature °F	0	35	80
Delivery Temperature °F	200	160	160
Max. Heat Load, BTU/HR	80,000	200,000	200,000
Working Fluid	R12	Water	R21
Required Flowrate, LB/HR	1300	1600	9800
Application	Manned or unmanned scientific payload with cold storage requirement	Manned Payload	Unmanned Scientific Payload
Heat Load Range	100:1	1000:1	1000:1

Physical Characteristics : Mechanical interface rather than fluid if possible.

On-Orbit Mate-up : Attachment to and deployment from the spacelab

SHRM

Controls : Applicable to any design return temperature

Working Fluid : R21 for flight system; R12 is allowable for prototype system

Envelope : Maximum diameter of 15 ft. for stowage in Shuttle cargo bay. Use volume above space lab pallet if possible.

External Environment : Design Conditions:

EARTH ORBIT	TRANSLUNAR	LUNAR ORBIT
. 200 n.m. full sun orbit	. fixed orientation	. 50 n.m. full sun orbit
. free-flying or attached to shuttle	. free-flying	. free flying

EARTH ORBIT	TRANSLUNAR	LUNAR ORBIT
. Radiant interchange with shuttle: none with other vehicles or solar cells	. no radiant exchange with other vehicles or solar cells	. no radiant exchange with other vehicles or solar cells

Launch and Re-entry Environment : Inside vehicle, so no aerodynamic heating thermal protection is required*. Use shuttle orbiter vibroacoustic environment

Design Factors of Safety

Structure : yield = 1.15 times maximum stress
ultimate = 1.5 times maximum stress
NOTE: Maximum stress is based on orbital condition or ground deployment conditions, whichever is worse.

Fluid Containing Components : Maximum expected operating pressure (MEOP)= working fluid vapor pressure at maximum expected system temperature

Proof Pressure = 1.5 times MEOP (The proof pressure should not result in stresses above the material's yield stress level.)

Burst Pressure = 2.0 times MEOP (The burst pressure should not produce stresses greater than the material's ultimate stress level.)

Fluid Lines : MEOP = Same as for components
Proof Pressure = 2.0 times MEOP
Burst Pressure = 4.0 times MEOP

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*Protection or accommodation equivalent to the usual shuttle payload bay or booster heat shield internal environment of 200-250°F will be considered. This requirement may set system operating pressure.

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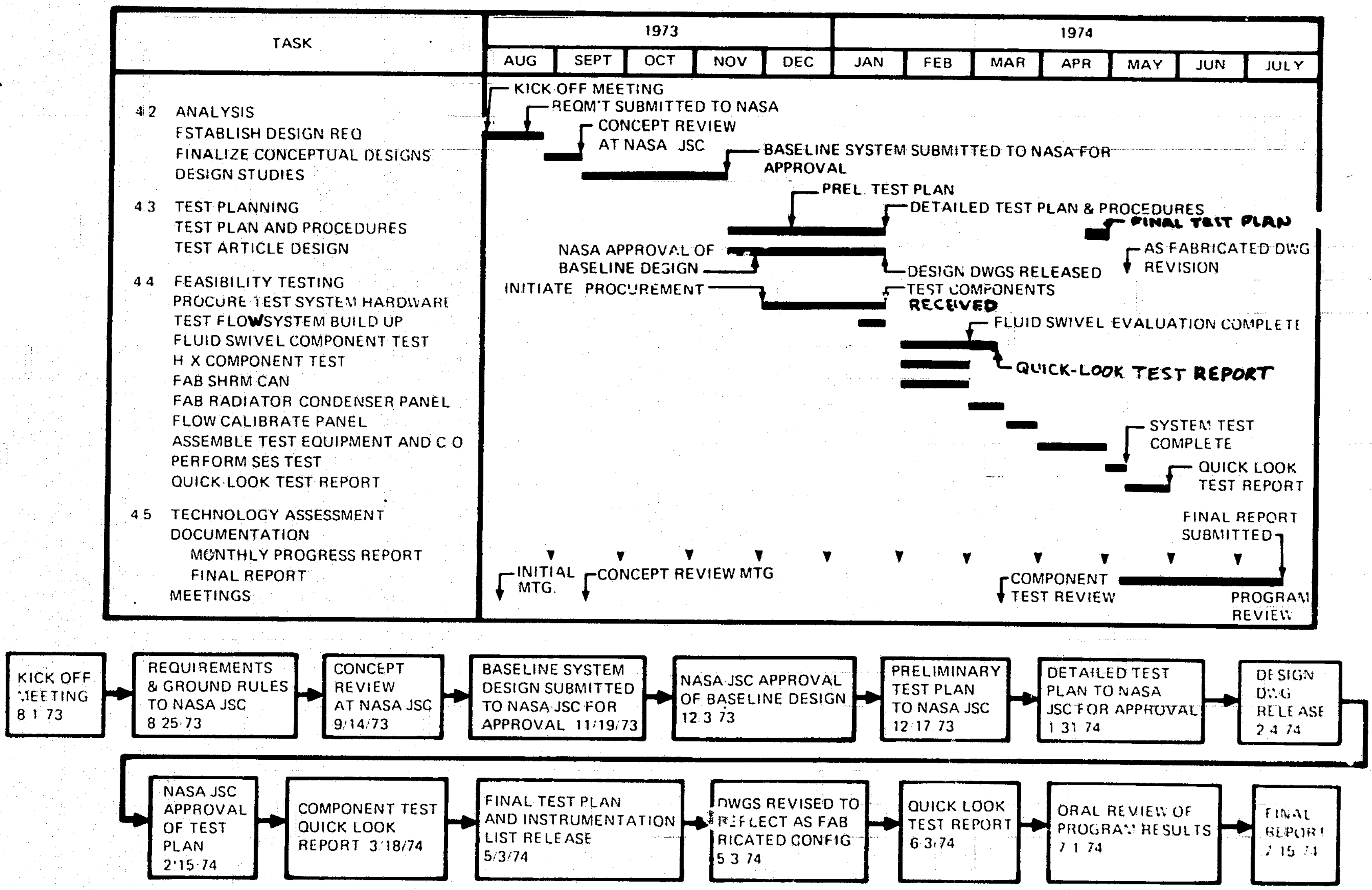

FIGURE 1 SELF-CONTAINED HEAT REJECTION MODULE PROGRAM SCHEDULE

(f) Structural Analyses - Perform stress and weight optimization analyses on the various deployment concepts.

(g) System Thermal Performance - Evaluate the overall thermal performance for each candidate system. Include an evaluation of control systems in this study.

(h) System Weight Analysis - Determine the total equivalent weight for each candidate system. (Equivalent weight includes hardware and support structure weight, fluid weight and power penalty.) Also establish the weight penalty associated with the flexibility provided by the modular nature of the SHRM.

(i) System Weight Optimization - Some effort will be made on the systems level; however, the bulk of the effort will be on the component level.

The planned analyses will be performed using existing design data to the greatest extent possible. Detailed computer models of components will not be used extensively in this effort. Some computer usage may be required to achieve results with sufficient accuracy; however, the short duration of the analysis effort prohibits the detailed modeling of all concepts. Documentation and references will be provided on all analyses conducted.

3.0 CURRENT PROBLEM AREAS

There are no current problem areas on this program.

4.0 WORK PLANNED FOR NEXT REPORTING PERIOD

The primary emphasis during the next reporting period will be on concept generation. An interim concept design review is scheduled for 14 September. The activity in support of this review will focus on generation of new concepts which were not included in the VSD proposal (Reference [1]).

5.0 REFERENCES

[1] Vought Systems Division Report No. 00.1599, "Technical Proposal - Development of A Self-Contained Heat Rejection Module", 2 May 1973.

APPENDIX B

DEVELOPMENT OF A SELF-CONTAINED
HEAT REJECTION MODULE

Contract No. NAS9-13533
Report No. T211-RP-002

PROGRESS REPORT
FOR THE PERIOD OF 29 AUGUST 1973 through 28 SEPTEMBER 1973

2 October 1973

Submitted by:

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORP.
DALLAS, TEXAS

TO

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
Houston, Texas

Prepared by:

J. L. Williams
J. L. Williams

Approved by:

R. J. French
R. J. French, Supervisor
EC/LS Group

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1.0 INTRODUCTION AND SUMMARY

This second progress report on Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 29 August 1973 through 1 October 1973.

Effort during this period was primarily concerned with concept selection and preliminary design of the prototype system. A meeting was held at NASA-JSC on 14 September 1973 at which VSD presented the Self-Contained Heat Rejection Module (SCHRM) concepts considered, and the concept recommended by VSD. The recommended concept is a sissor-type deployment mechanism, which is similar in size and operating characteristics to the Apollo Telescope Mount (ATM) solar cell assembly deployment mechanism; which was used successfully on Skylab. Analysis efforts have centered on sizing of the prototype SCHRM which is to be constructed later in the program.

Work planned for the next reporting period includes establishment of the prototype SCHRM configuration, and completion of design analyses.

2.0 DISCUSSION OF WORK PERFORMED IN THIS REPORTING PERIOD

2.1 Radiator Deployment Concepts

At NASA's request (Reference [1]), the radiator deployment concepts presented in the VSD Technical Proposal (Reference [2]) were expanded to consider SCHRM use in the shuttle orbiter cargo bay, and to consider higher heat loads. The results of this effort are presented in Appendix A, which is the presentation given to NASA-JSC on 14 September with corrections and references added. Appendix A illustrates all of the approaches considered.

The VSD recommended approach, which is illustrated in Figure 1, involves the use of a sissor mechanism to deploy the radiator panels. It is shown in Figure 1, for free-flying space station and shuttle orbiter cargo bay application. This approach is similar to that used for the Apollo Telescope Mount (ATM) Solar Cell Array. For SCHRM, the sissor deployment mechanism permits a concept which has the advantages that it:

(1) has low technical risk, since the deployment mechanism was used successfully on Skylab,

(2) is equally applicable to free-flying space stations, or to the shuttle orbiter cargo bay,

(3) permits a docking port or a transfer tunnel to be incorporated with the SCHRM,

(4) has low fabrication costs since flat, rectangular (or square) radiator panels are used,

(5) can accommodate a variety of manifolding techniques,

(6) incorporates fluid manifolds as an integral part of the deployment system, thus reducing weight, and

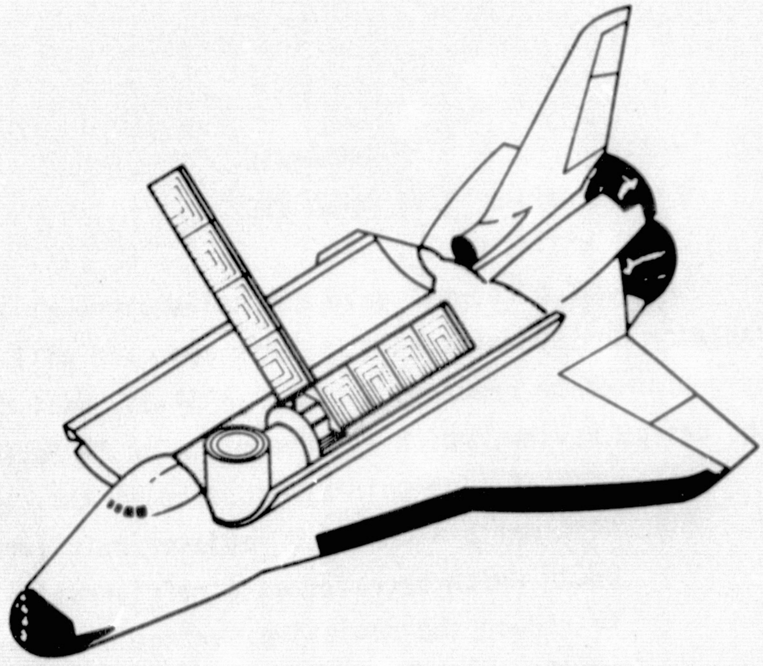
(7) can easily incorporate a radiator retraction capability.

In addition, the recommended approach has a competitive volume requirement.

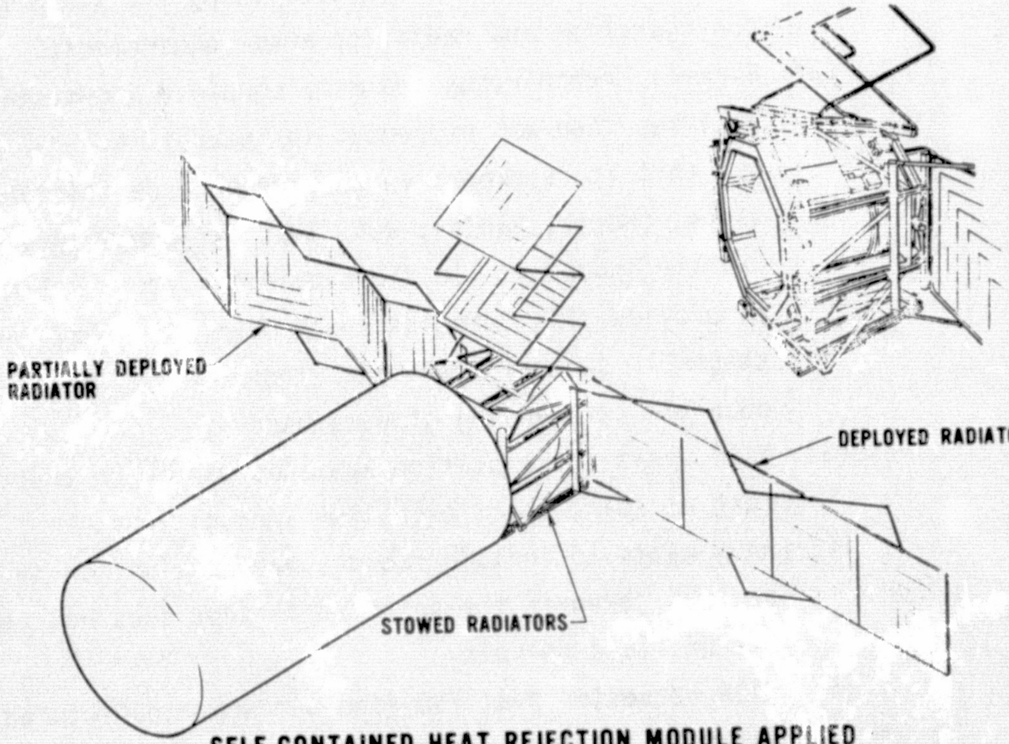
2.2 Concept Review Meeting

The concept review meeting was held at NASA-JSC on 14 September 1973. Participants were:

Wil Ellis	NASA-JSC
Burrell French	NASA-JSC



SELF-CONTAINED HEAT REJECTION MODULE (SHRM) APPLIED TO EXPERIMENT MODULE IN THE SHUTTLE ORBITER CARGO BAY



SELF-CONTAINED HEAT REJECTION MODULE APPLIED TO A FREE-FLYING EXPERIMENT MODULE

FIGURE 1 - SELECTED RADIATOR DEPLOYMENT CONCEPT

Walt Guy	NASA-JSC
Keith Hudkins	NASA-JSC
J. L. Williams	VSD

NASA-JSC personnel gave tentative approval to the recommended SCHRM radiator deployment concept. This approach will be used in the preliminary design to be rendered by VSD. NASA-JSC will reserve final judgment until the design review, which is scheduled for 19 November 1973.

Other significant points discussed in the meeting were:

(1) NASA wants a combined radiator/refrigeration unit in the SHCRM which operates as a refrigeration unit as required to reduce the radiating surface area requirment. Reference [1] implies that SCHRM would be capable of operating in either a refrigeration or radiator mode, but that the operating mode would be exclusive on any given mission. Appendix A also reflects this approach. VSD was directed to consider the dual-mode system in establishing the SCHRM flow loop configuration and radiating area requirements.

(2) Burrell French requested more complete documentation on analyses than was presented in Appendix A. VSD pointed out that it is expensive to reproduce definitions, derivations, charts, tables, etc. that are commonly used throughout the industry. In VSD's opinion, it is not necessary to provide this information to make the study results understandable several years into the future. VSD will provide complete referencing of data used.

(3) NASA wants space station applications to be emphasized as much as space shuttle orbiter applications.

(4) NASA wants to include fluid swivel evaluation in the test program, even if the selected concept does not require the use of fluid swivels.

(5) NASA requested that VSD find out who manufactured the ATM Solar Cell Array so NASA can obtain documentation on the deployment mechanism, and can attempt to obtain spare hardware for use in the SCHRM.

2.3 ATM Solar Cell Array Deployment Mechanism

VSD contacted NASA-MSFC and found that the solar cell array was developed in-house. John Calvert of the MSFC S&E Division said the array was documented in "Solar Array Mechanical System Development Report", which has no identifying number.

Burrell French ordered the report, and arranged for a visit to MSFC on 2 October 1973 by VSD and MSC personnel. The purpose of the visit is to inspect the array and deployment mechanism, and, if it is feasible to use the mechanism in SCHRM, to obtain a spare unit if possible.

2.4 Radiator/Refrigerator System Design

2.4.1 Radiator Area

The ground rules for this study, as documented in Reference [1], are shown in Figure 2. The radiator required for this application is sized as follows:

(1) Break the radiator into 5 zones of equal heat rejection

(2) Calculate fluid inlet and outlet temperatures for each zone,

$$T^i_{out} = T^i_{in} - \frac{q^i}{\dot{m}\,Cp}$$

Where

T^i_{out} = outlet temperature for zone i

T^i_{in} = inlet temperature for zone i

q^i = heat rejection from zone i

$\dot{m}$ = radiator flowrate

Cp = radiator fluid specific heat

(3) Calculate the average radiating temperature for each zone

$$T^i_{av} = (T^i_{in} + T^i_{out})/2 - \Delta T_{t-f}$$

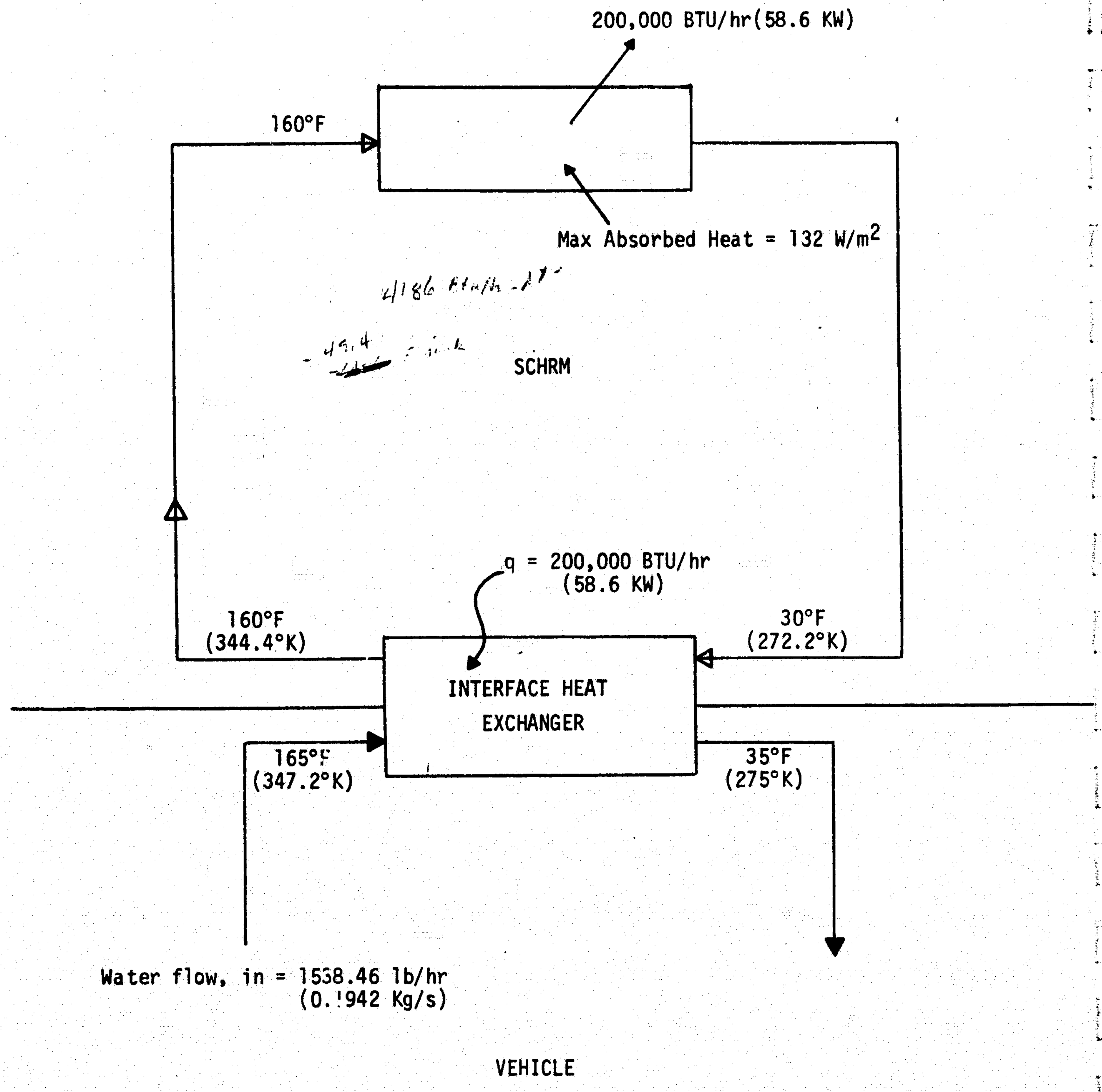


FIGURE 2 SCHRM INTERFACE HEAT TRANSFER REQUIREMENTS

Where

T_{av}^{i} = zone average temperature

ΔT_{t-f} = fluid-to-tube temperature drop (5.5°K for single-phase radiator and 3°K for condensing radiator; these values are typical of halocarbon fluids in space radiators)

(4) Calculate heat rejection per unit area from each zone

$$\left(\frac{q}{A}\right)_i = \sigma\varepsilon\eta(T_{av}^{i})^4 - \left(\frac{q}{A}\right)_a$$

σ = Stefan-Boltzman constant, 5.67×10^{-8} W/m^{2}K^4

ε = radiator emittance (0.8 for silver teflon)

η = radiator fin effectiveness (assumed to be 0.9)

Figure 3 presents heat rejection vs radiator temperature for absorbed heat fluxes of 15 w/m^2 and 132 w/m^2.

(5) Calculate zone area

$$A_i = \frac{q_i}{q/A}$$

(6) Calculate total area

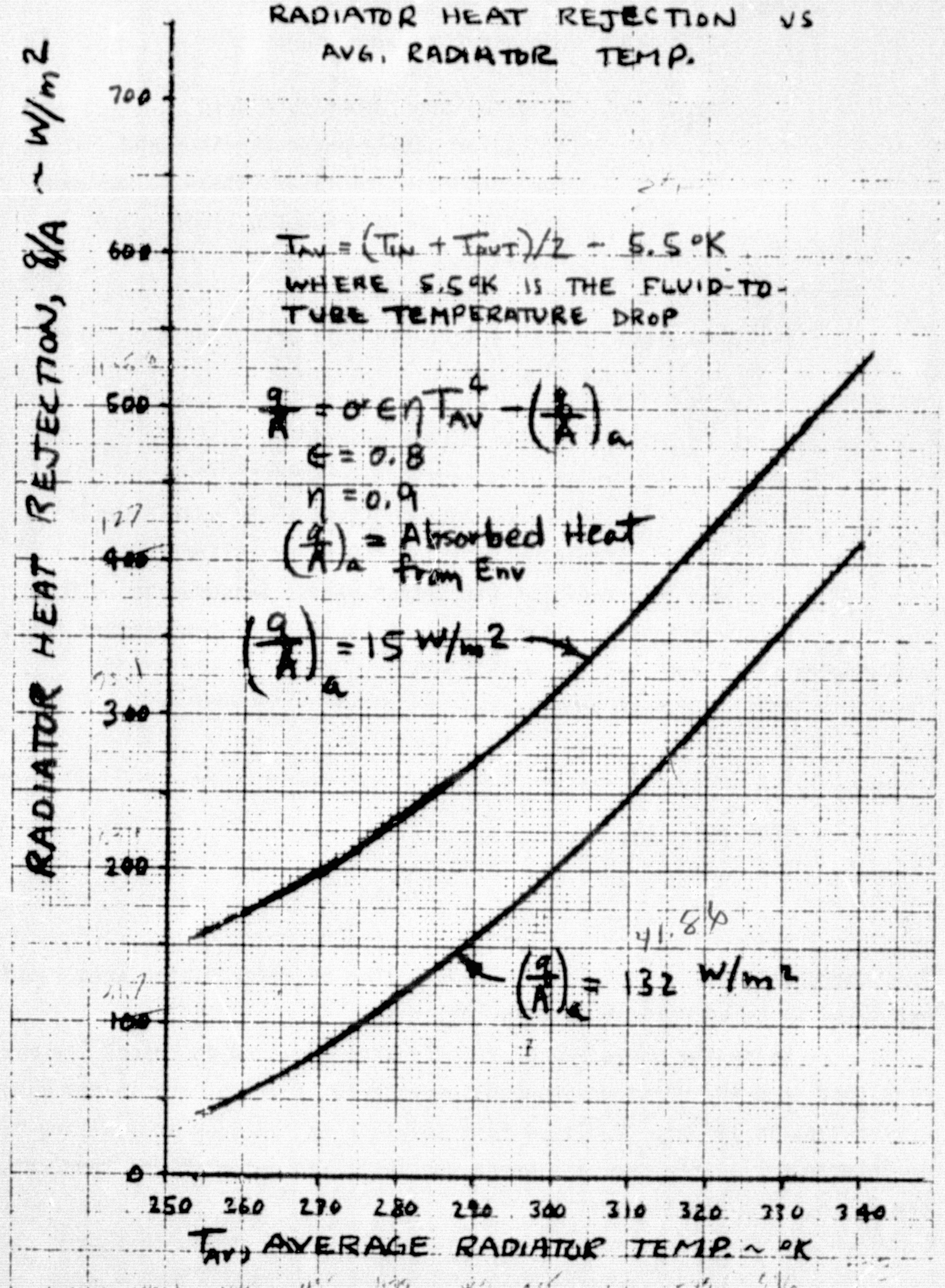
$$A_T = \sum_{i=1}^{5} A_i$$

This approach gives a reasonable approximation to the radiator area required, estimated to be accurate to within 10%.

Using the above procedure, the area required to reject the maximum heat load into the worst environment considered is 325 m^2 and in the minimum environment is 187 m^2. A closed form solution to radiator area sizing is given in Reference [11] for the case where the inlet and outlet fluid temperatures are known. The equation is

FIGURE 3

RADIATOR HEAT REJECTION VS AVG. RADIATOR TEMP.



$$A = \left[\frac{1}{4} \ln \left(\frac{\tau_2 + 1}{\tau_2 - 1}\right) + \frac{1}{2} \tan^{-1} \tau_2 - \frac{1}{4} \ln \left(\frac{\tau_1 + 1}{\tau_2 - 1}\right) - \frac{1}{2} \tan^{-1} \tau_1\right] \left[\frac{\dot{m}Cp}{\sigma \varepsilon \eta_o T_s^3}\right]$$

Where:

$\tau_1 = T_{in}/T_s$

$\tau_2 = T_{out}/T_s$

T_s = equivalent sink temperature $= \left[\frac{(q/A)a}{\sigma\varepsilon}\right]^{1/4}$

T_{in} and T_{out} should be reduced by the temperature drop between the fluid and tube to obtain realistic results. This approach gives an area requirement of about 300 m^2 for the maximum absorbed heat environment and 180 m^2 for the minimum absorbed heat environment. These numbers are similar to those obtained above though somewhat smaller. The larger values will be used, since they are more conservative. The selected concept would require deployment of 18 panels of 3m x 3m dimensions to achieve an area of 325 m^2, considering radiation from both sides. It is possible to use vapor cycle refrigeration to reduce this area requirement.

2.4.2 Refrigeration System Design

The first step in evaluation of a refrigeration system is analysis of the cycle. The efficiency of refrigeration systems is usually expressed in terms of Coefficient of Performance (COP). The best possible performance which can be achieved is (References [4] and [8]):

$$\text{Carnot COP} = \frac{T_L}{T_H - T_L}$$

Where:

T_L = temperature at which heat is absorbed

T_H = temperature at which heat is rejected

The best performance which can be obtained with a real fluid is less than this because of inherent losses associated with throttling and with superheating during compression. Figure 4 shows an ideal cycle using Refrigerant 12 (R12) as the working fluid. Starting at (a), the fluid is vaporized in the evaporator to point (b). It is then compressed to point (c), and then is desuperheated and condensed to point (d). It is then expanded through a throttling valve to point (a), thus completing the cycle. The COP for this cycle is

$$COP = \frac{\text{Cooling Effect}}{\text{Work}} = \frac{h_b - h_a}{h_c - h_b}$$

Frequently an efficiency, E_c, which compares this COP to carnot COP is used to evaluate fluids (Reference [5]).

$$E_c = \frac{COP}{\text{Carnot COP}}$$

E_c ranges from 0.8 to 0.9 for most refrigerants, and is about 0.85 for R-12.

The COP defined above cannot be achieved in an actual system because of inefficiencies in the compressor and compressor drive motor, and pressure drop in the condenser and evaporator. Once these inefficiencies and pressure drops are defined, actual COP can be computed. Figure 5 shows actual COP vs condenser temperature for a 272.2°K evaporator for the indicated conditions. In order to determine how reasonable the results were, a new efficiency E_a was defined:

$$E_a = \frac{\text{Actual COP}}{COP}$$

For the system in Reference [3], which uses R-12 and a heli-rotor compressor, E_a was found to be 0.59 at a condenser temperature of 313.9°K and an evaporator temperature of 272.2°K. Figure 5 has a value of E_a of 0.56 for similar conditions, so it reflects lower performance than can be obtained with an existing system.

Figure 5 can be used in conjunction with Figure 6 to calculate the required radiator area and power requirement for a spacecraft refrigeration system. Figure 6 shows the heat rejection from the condensing radiator as a function of condenser temperature. This curve ignores the superheated and

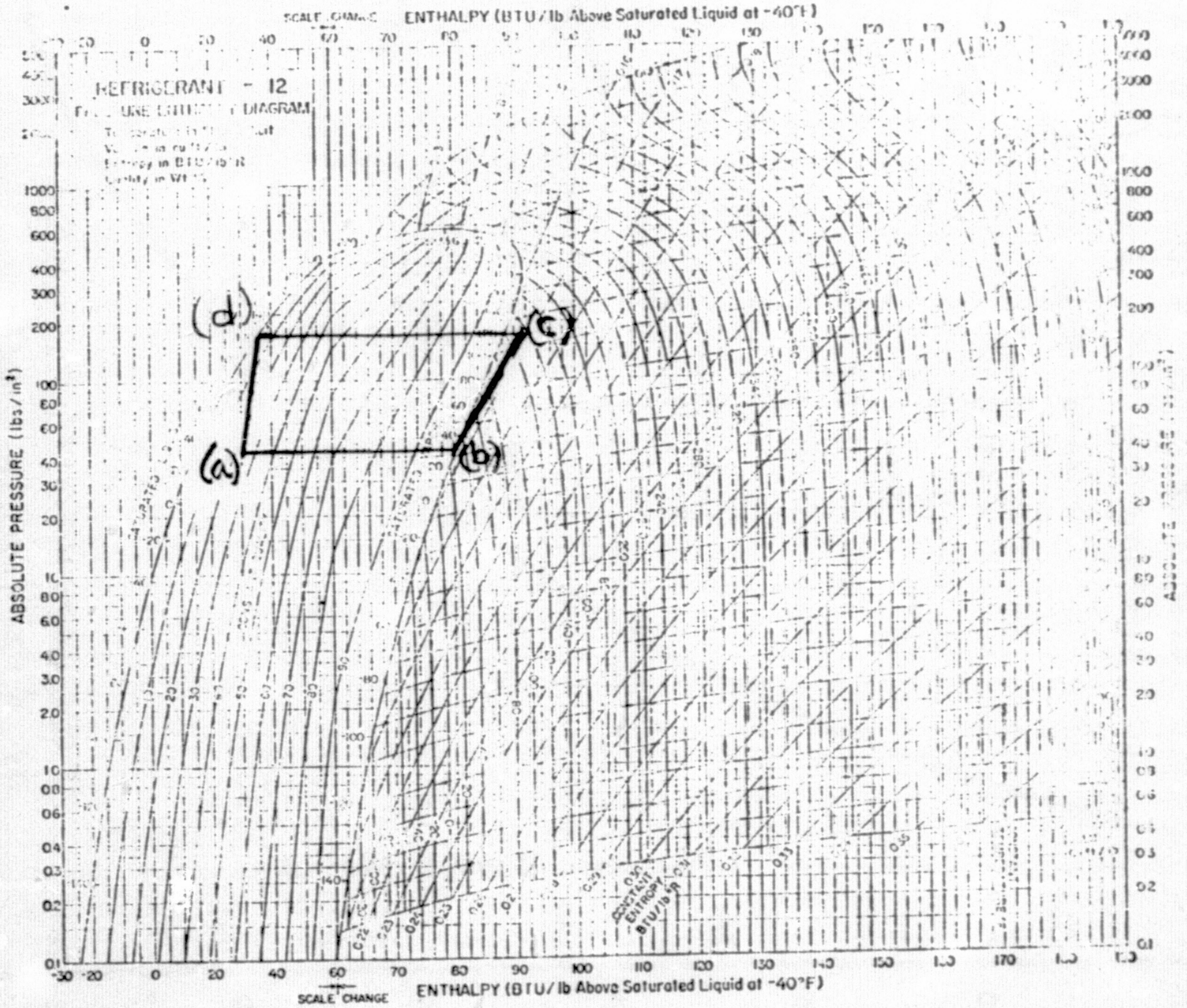


FIGURE 4 IDEAL REFRIGERATION CYCLE WITH R-12 AS THE WORKING FLUID

FIGURE 5

ACTUAL COP VS. CONDENSER TEMPERATURE

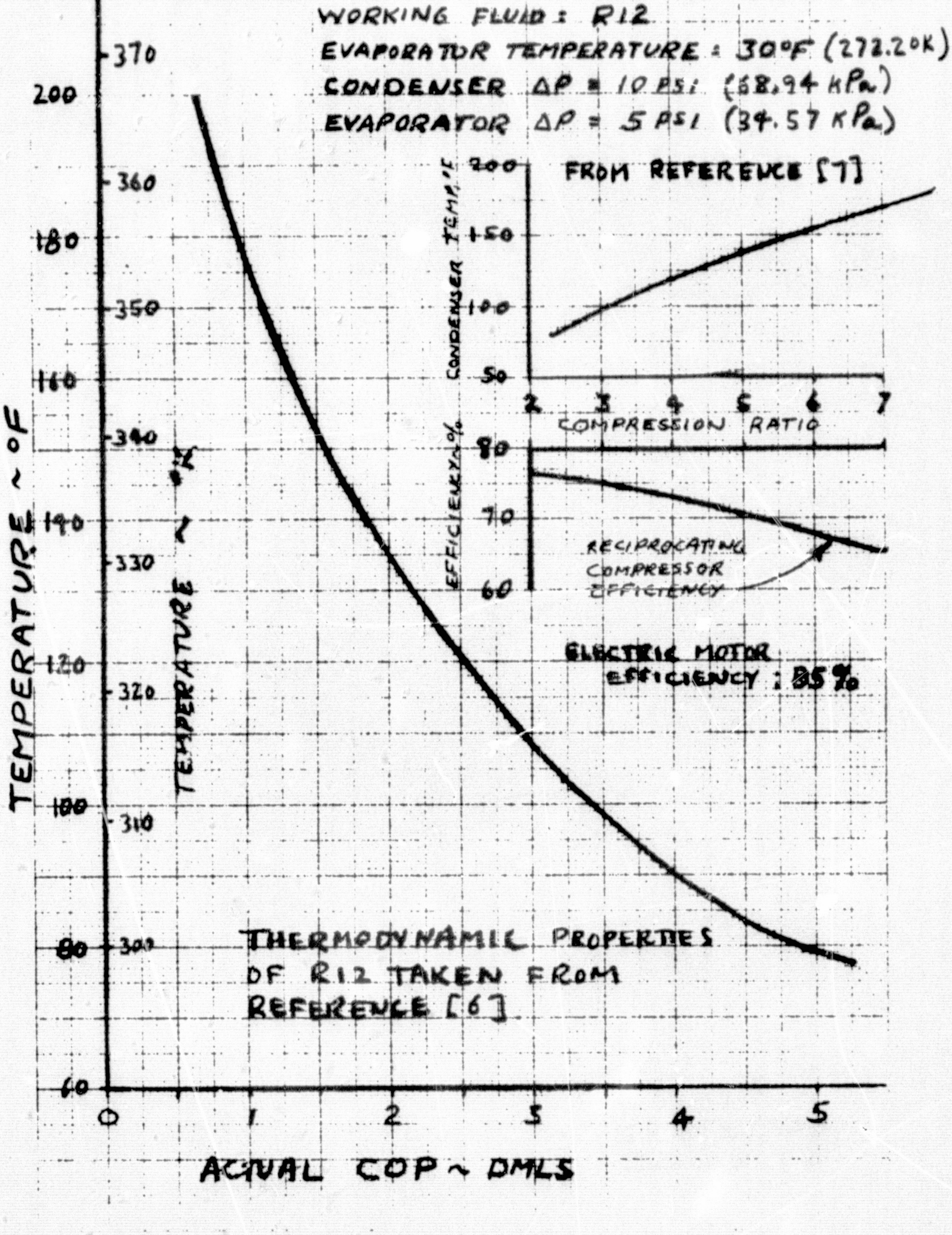
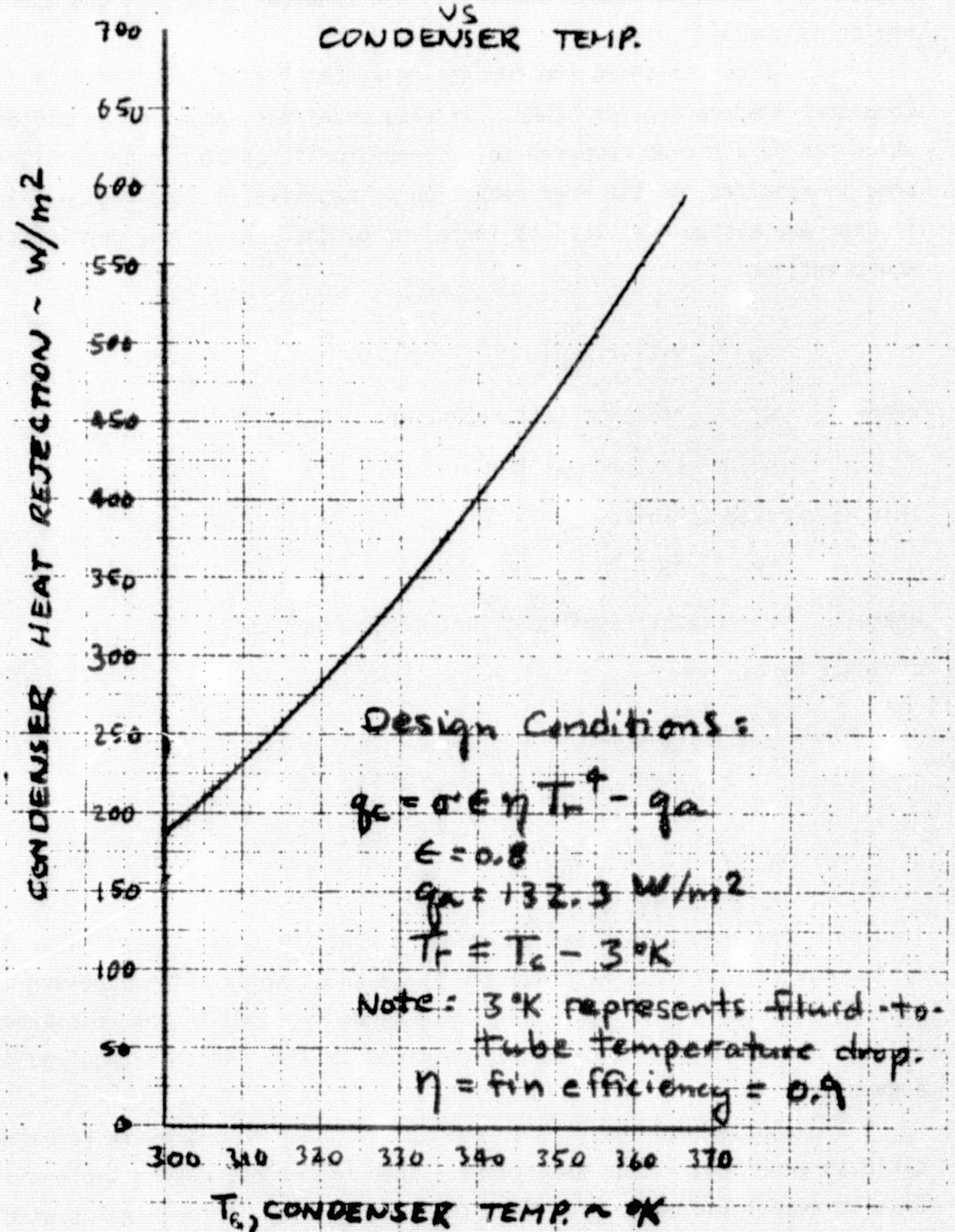


FIGURE 6

RADIATOR HEAT REJECTION VS CONDENSER TEMP.



JLW
10/2/73

subcooled regions of the condenser, which is conservative since the superheated region is at a higher temperature than the condensing region, and the sub-cooled region is small.

Figure 7 shows the condensing radiator area required as a function of condenser temperature for SCHRM. It also shows the reduction in radiator area which can be achieved compared to a conventional radiator. Below 310°K more area is required, so the area reduction is negative in that region. In this figure, the system heat load is increased by the work of the refrigeration equipment, or

$$q_r = q_1(1 + \frac{1}{COP})$$

Where: q_r = radiator heat rejection

q_1 = system heat load

This is derived as follows:

$$q_r = q_1 + w$$

Where: w = work required to drive the compressor

dividing by q_1

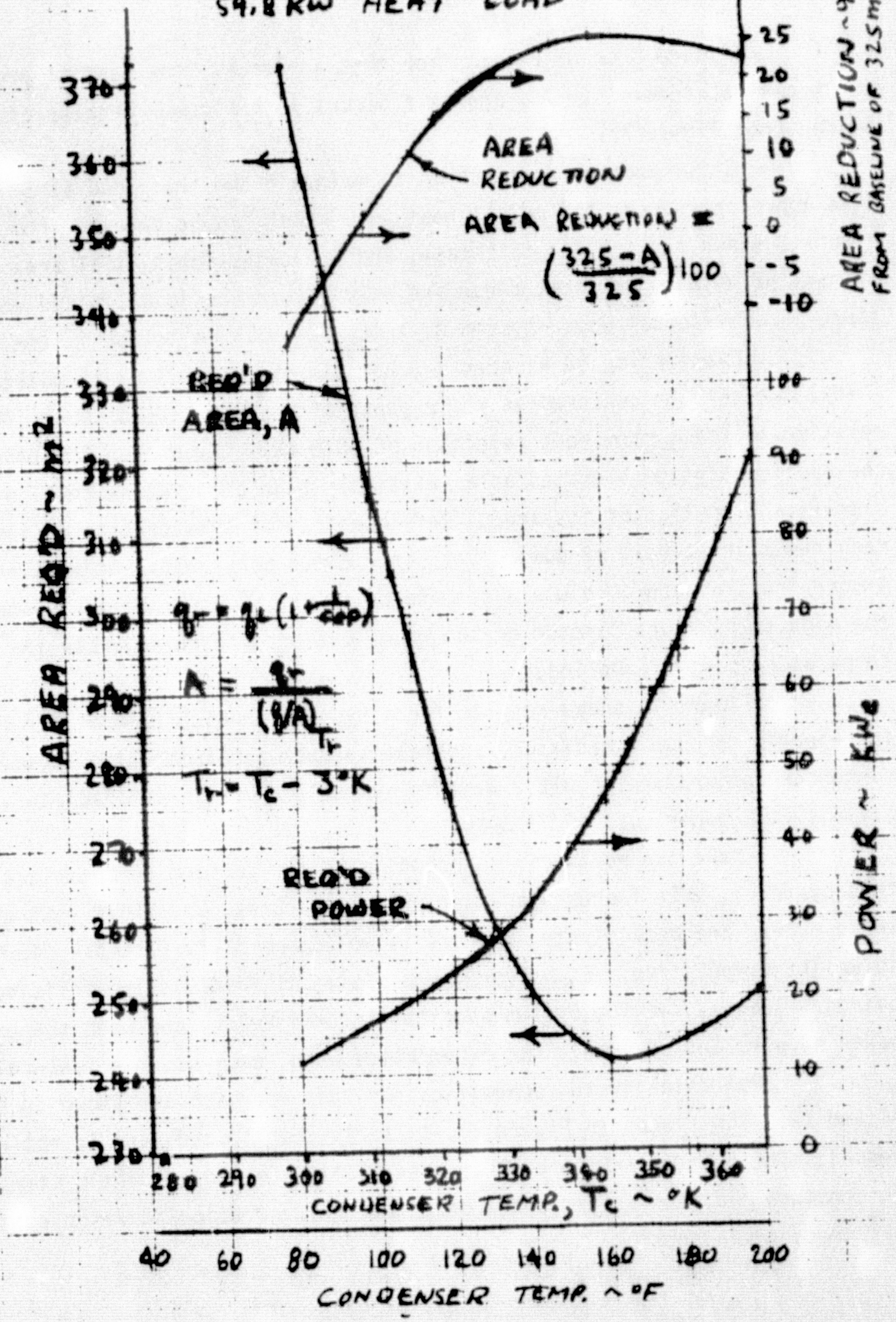
$$\frac{q_r}{q_1} = 1 + \frac{w}{q_1} = 1 + \frac{1}{COP}$$

$$q_r = q_1 (1 + \frac{1}{COP})$$

This is a reasonable approach because the power to drive the refrigeration system may come from an external source (such as solar cells); however, the work of the compression cycle will have to be rejected in the refrigeration cycle. The maximum area reduction obtained is 25%, and that required 45KW to accomplish.

Another approach is to assume the power to drive the refrigeration cycle is generated within the system. Then the refrigeration cycle will also have to reject the waste heat from the power cycle. A fuel cell system is assumed, with an efficiency, e, of 62.5%. The heat rejected from the refrigeration system is then:

FIGURE 7
RADIATOR AREA VS. CONDENSING TEMP.
COMPRESSOR WASTE HEAT ADDED TO
59.8 KW HEAT LOAD



$$q_r = q_1 \left(1 + \frac{1}{e[COP]}\right) = q_1 \left(1 + \frac{1.6}{COP}\right)$$

Figure 8 shows results for this approach. The largest area reduction which can be achieved is 7%, and this occurs with a power penalty of about 30 KW (from Figure 7).

Still another approach is to maintain the heat load at 58.6 KW. This approach reduces the actual heat load which can be removed from the system. Figure 9 shows results for this case, and indicates substantial area savings. It must be pointed out that these are only obtainable to a limited degree unless there is an external power system with its own heat rejection system.

Examination of Figures 7-9 reveals that significant radiator savings (>40%) can only be achieved at the expense of a great deal of power (>25 KW) relative to the system heat rejection of 58.6 KW. The reason for this is that the radiator system operates at a relatively high temperature level over a significant portion of its area. Refrigeration of this portion of the heat load requires a great deal of power, yet contributes no significant area reduction. A more fruitful approach would seem to be to use a pure radiator system for the high temperature portion of the heat load, and to refrigerate the lower temperature part of the load.

Figure 10 shows results for a system in which half of the heat load is rejected through a radiator, and half through a refrigeration system. At a condenser temperature of 322.2°K, this system produces a 40% radiator area savings at a power cost of 11.5 KW.

There are no specific ground rules for performing optimization of this system. Table I presents a set of ground rules, and shows that under those rules, the above system appears to be economically feasible, in that it is weight competitive. It is very competitive if installed power is not required. That is, if the system operates intermittently and uses the existing power supply, and pays only for expendables used, such as in a fuel-cell system.

Potential system schematics for this approach are shown in Figures 11 and 12. The system in Figure 11 is fairly complicated, and is not recommended. The system in Figure 12 is essentially two independent systems, a high temperature radiator system and a dual mode refrigeration/radiator system. It has these advantages:

FIGURE 8

RADIATOR AREA VS CONDENSER TEMP.
HEAT LOAD INCLUDES COMPRESSOR
WASTE HEAT, AND WASTE HEAT
FOR ADDITIONAL POWER TO DRIVE
COMPRESSOR

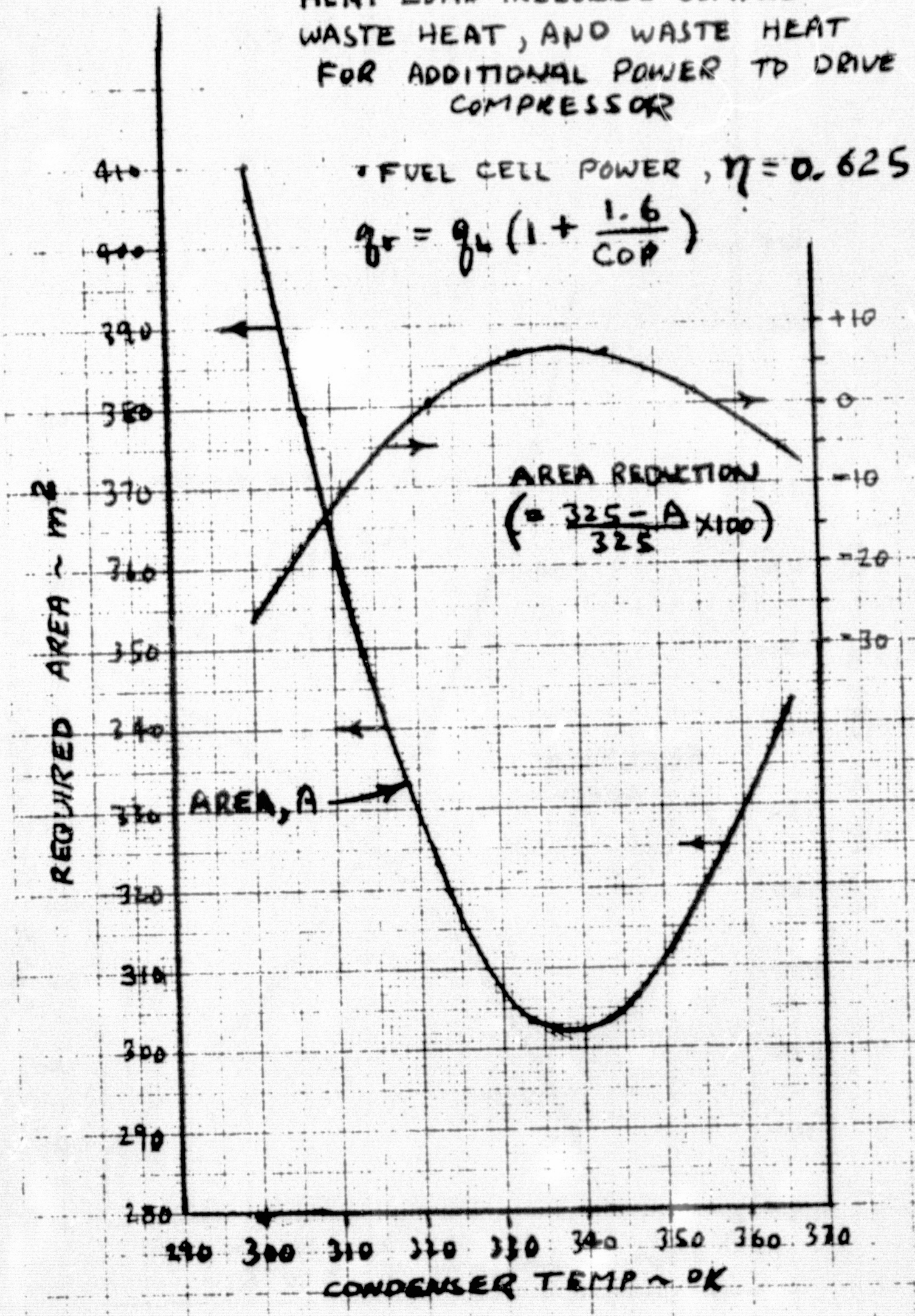


FIGURE 9
RADIATOR AREA VS CONDENSER TEMP.

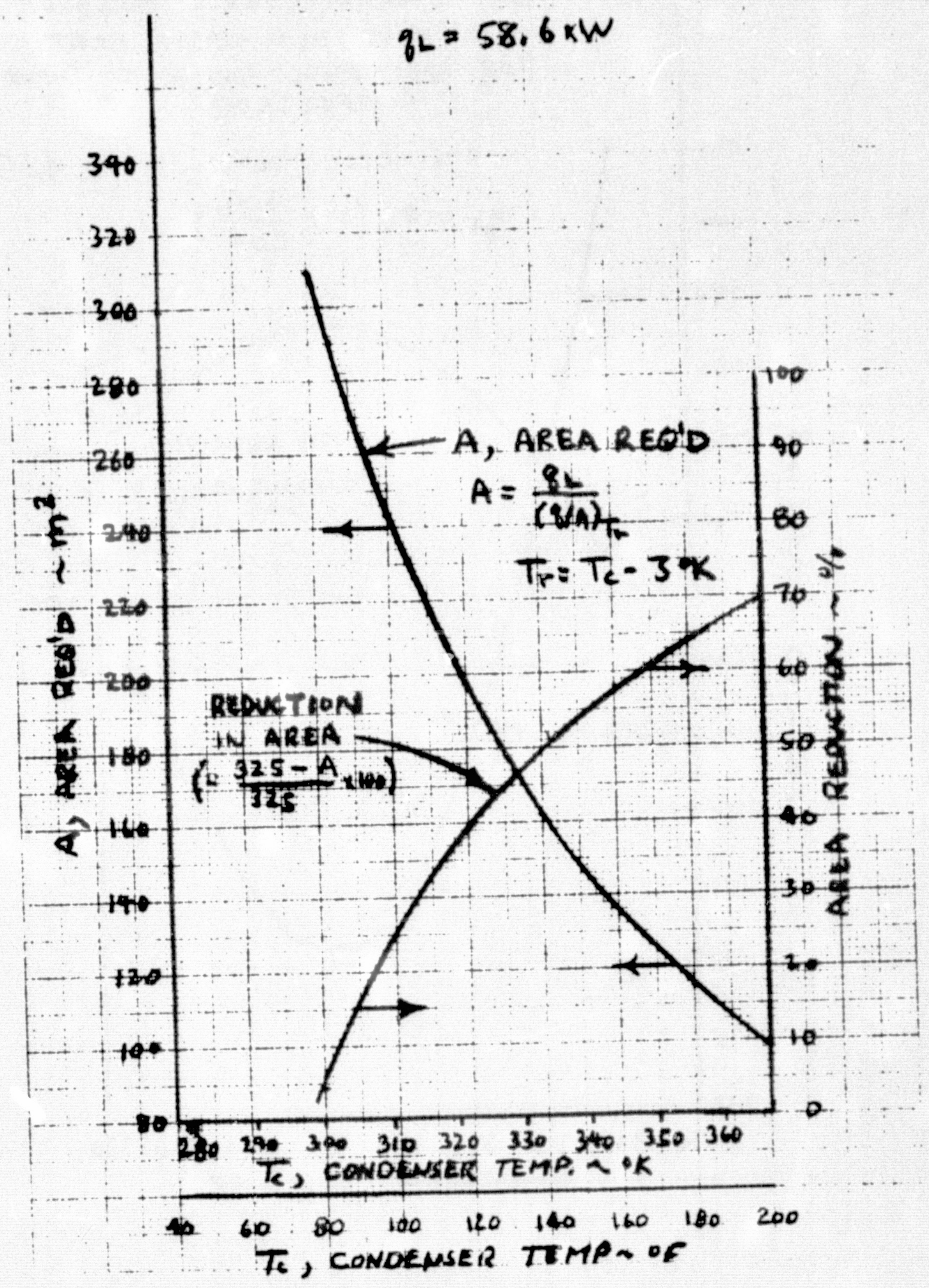


FIGURE 10

RADIATOR AREA VS CONDENSER TEMP

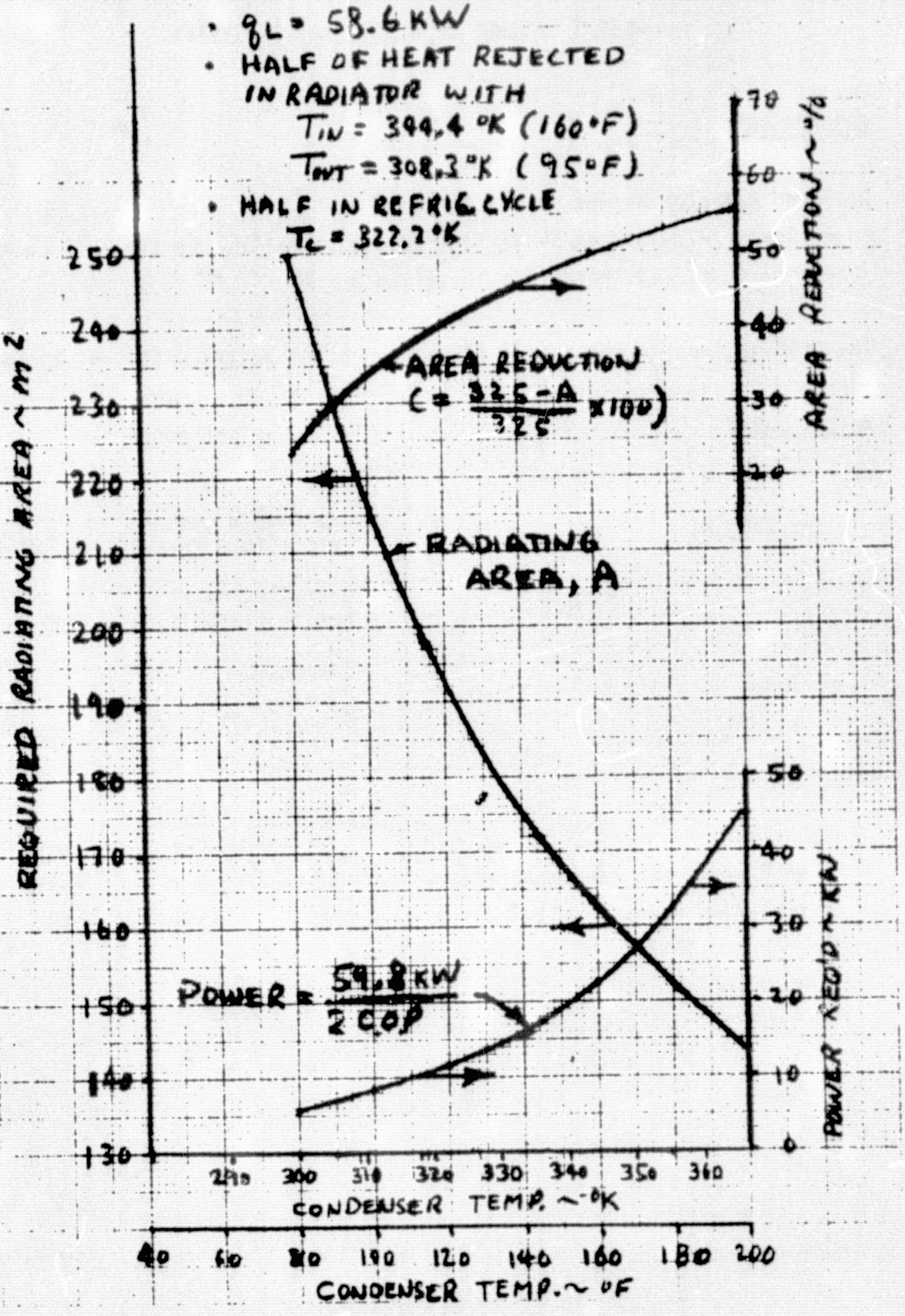


TABLE I

WEIGHT SAVINGS OF REFRIGERATION SYSTEM

GROUNDRULES:

Deployed Radiator Weight = 12.25 kg/m^2

Power Cost = 13.9 Kg/KW if fuel cell expendables are the only cost

= 139 Kg/KW for installed power

The area savings of the system (from Figure 10) is 325 - 194 = 131 m^2

Weight Saved

Wt. = (131 m^2)(12.25 Kg/m^2)

= 1605 Kg

Weight Added

Power -

Expendable system
wt. =(13.9 Kg/KW)(11.5 KW) = 160 Kg

Installed Power
wt. = (139)(11.5) = 1600 Kg

Compressor - 34 Kg

Misc. Equip.- 20 Kg

Weight Summary

	Saved Kg	Added Kg	Net Kg
No Power Installation Penalty	1605	214	-1391
Installed Power Penalty	1605	1654	+49

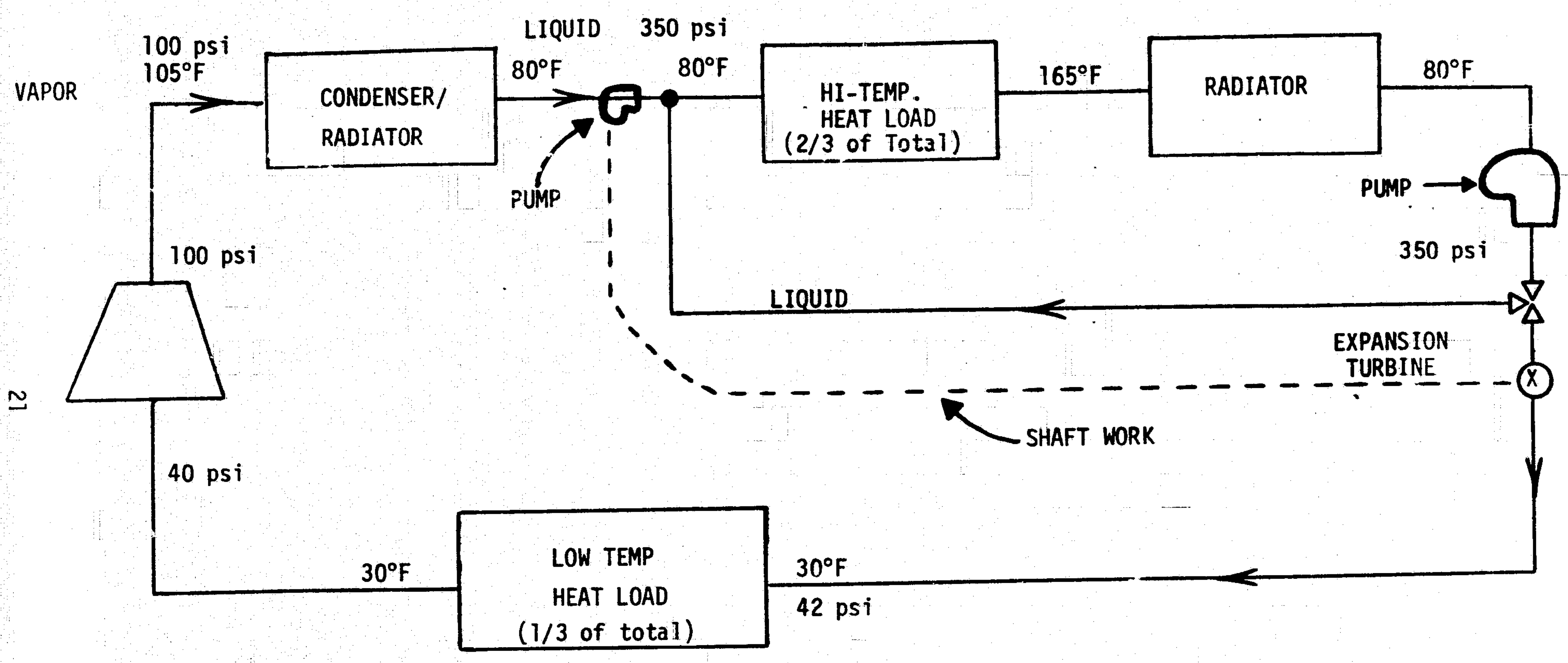


FIGURE 11 HYBRID REFRIGERATION SYSTEM

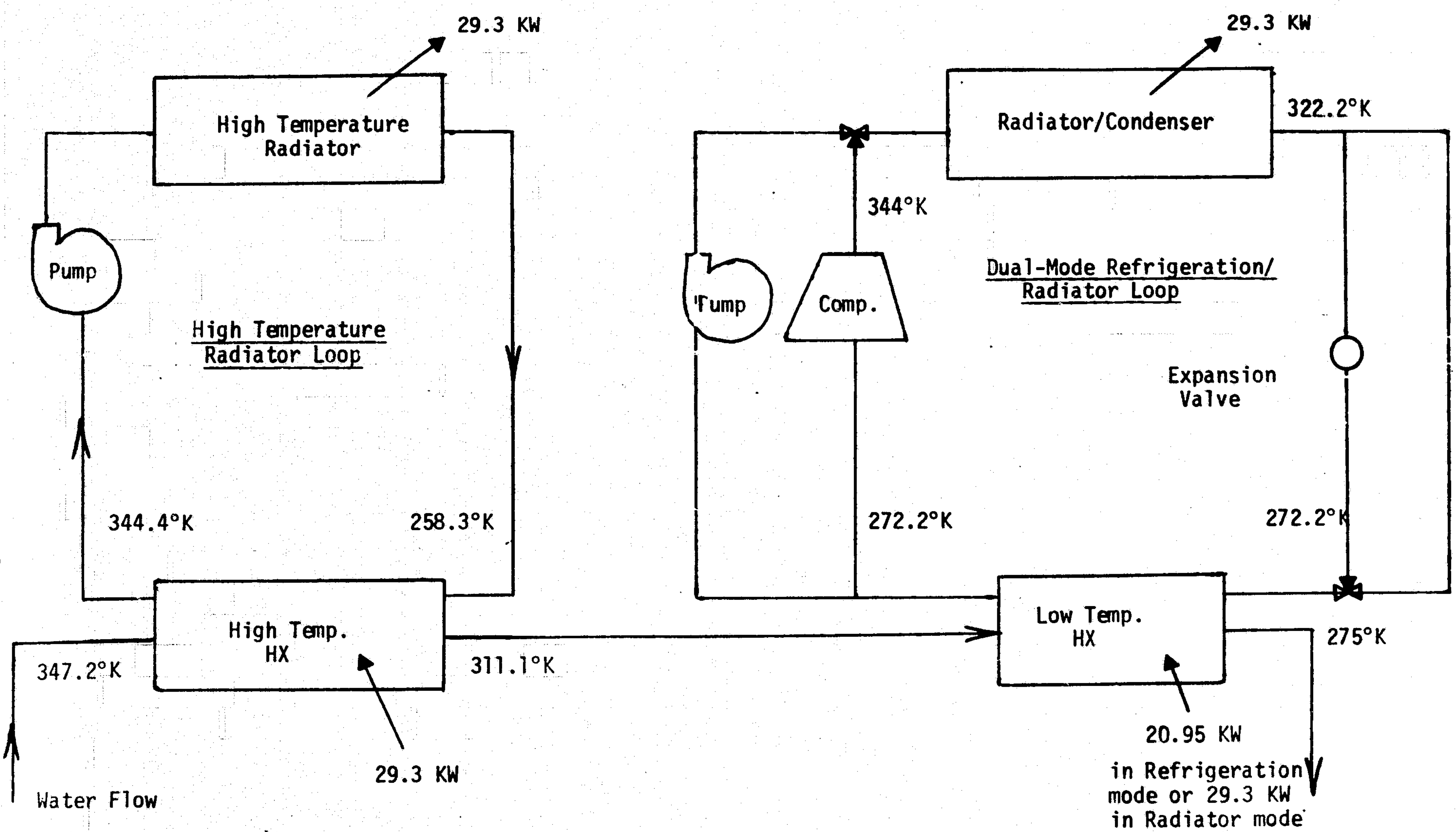


FIGURE 12 TWO SEPARATE HEAT TRANSFER LOOP CONFIGURATIONS

(1) Separate controls can be used for each system

(2) For some applications only half of the system would be required: for example, for very high temperature applications the high temperature radiator would be used; for 0°C evaporator only the refrigeration unit would be used.

(3) The total radiating area of 193 m^2 exceeds the area required for full-load operation in a favorable environment, which is 187 m^2.

(4) Parallel flow radiator panels can be used on the condensing radiator and two-dimensional tube layouts can be used on the high temperature radiator. The high temperature radiator would be used to achieve load control.

VSD recommends that SCHRM be designed to the above flow configuration concept, as depicted in Figure 12, for the reasons cited above. The system shown in Figure 12 has not been optimized, and cannot be optimized without specific trade parameters such as power availability, power cost, etc. Since these parameters are strongly vehicle oriented, and are thus not known at this time, accurate optimization is not possible. However, it seems likely that a reasonable split between the low temperature heat sink requirement (~30°F) and the higher temperature sink requirement (~80°F) can be defined. The lower temperature heat sink should only be necessary for crew comfort control and atmosphere revitalization requirements. For a four-man crew this should be limited to around 2 KW_t maximum. There could also be some navigation and guidance equipment which require low temperature cooling. This might raise the total low temperature load to 3-4 KW out of a total load of 58.6 KW. This assumes that other equipment could be cooled by fluid at 80-120°F. This should be possible, but it might call for special design provisions for the cold plates on that equipment. Under this load split the refrigeration system would draw only about 1.6 KW at the peak loading condition; but the radiator savings would be about the same. This approach would be very attractive from the power consumption standpoint. NASA-JSC guidance in this area is requested.

2.4.3 Radiator Panel Configuration

The radiator panel configuration is dependent on the testing philosophy which is adcpted in the program. The VSD thermal-vacuum chamber has interior dimensions of 3.048-m diameter by 3.048-m length. This severely limits the size of the radiator system which can be deployed in the chamber. There are two possibilities:

(1) large panels can be used, but with only partial deployment in the chamber, or

(2) small panels can be used with full deployment in the chamber.

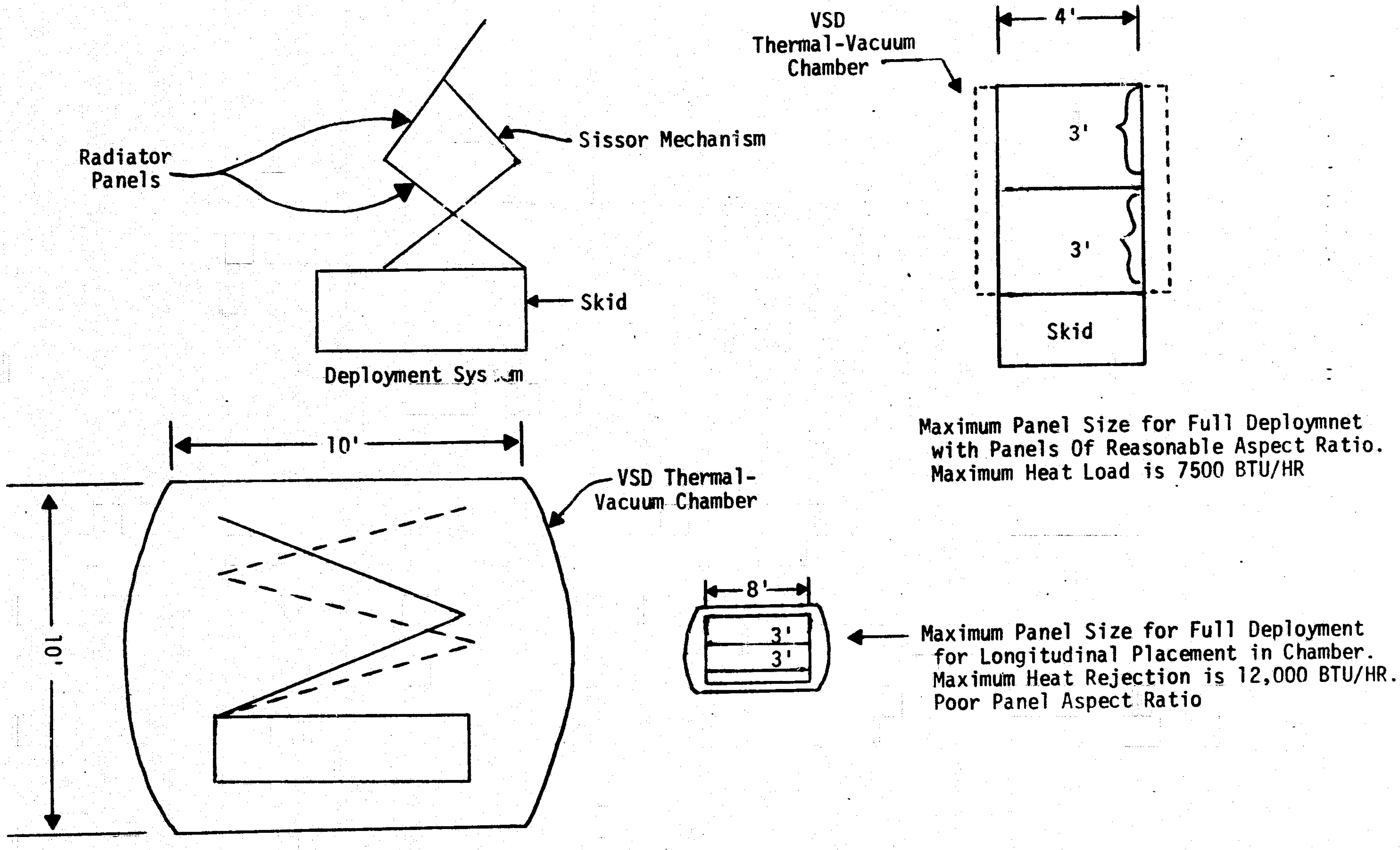
Neither approach is wholly satisfactory. If large panels are used they can be deployed as shown in Figure 13. The maximum system heat rejection is 3.5 KW. A problem with this approach is the gravity head which will develop in the panels since they are not horizontal. This is undesirable in thermal-vacuum tests of radiator systems. The small panel deployment approach is shown in Figure 14. This approach allows horizontal placement of the panels, but the available heat rejection is only 2 KW.

There is also the possibility that an ATM Solar Cell Array deployment mechanism can be obtained from NASA-MSFC. This would establish the radiator panel size and detail design.

The specific flow configuration and detail design of the radiator panels cannot be accomplished until after the testing approach is resolved.

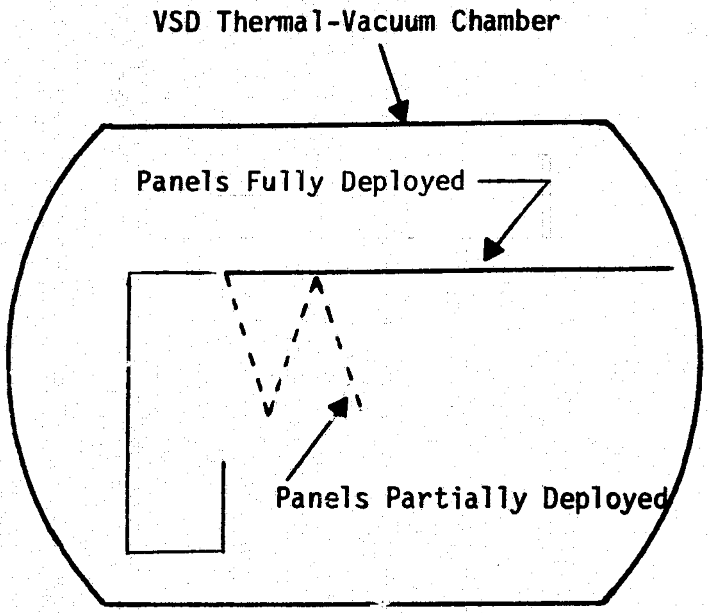
2.5 Proposed Fluid Swivel Fitting Test

The schematic shown in Figure 15 illustrates the test setup recommended for testing candidate fluid swivel fittings at the low temperature/vacuum environment required for the system. The swivel is cooled as required by the controlled flow of LN_2 into the line wrapped around the swivel body and inlet line. The outlet line from the swivel is rotated (cycled) by a torque motor which can be programmed as required. The fluid line is evacuated and filled with coolant (F-21 or other as required). Fluid pressure is controlled at the reservoir. Gross leakage down to 0.5 lb/hr can be measured with the flowmeter in the inlet line. Lower leakage rates can be detected by dumping the freon, filling the line with helium, and monitoring the mass spectrometer connected to the chamber. The VSD SES satellite chamber is approximately 4' dia by 6' long, with the door plexiglass for viewing the test article, and



3 Panels 8' x 5' Each In Chamber -
Maximum Possible Extent of Deployment Shown - Max. Heat Rejection = 11000 BTU/HR

FIGURE 13 POSSIBLE METHODS OF SCHRM DEPLOYMENT IN THE VSD THERMAL-VACUUM CHAMBER



DEPLOYMENT SCHEME

Panels are 3' by 5'
Max A = (9)(5)(2) = 90 ft^2
q = (q/A)A = 11,250 B/H

Proper aspect ratio panels -
3' high and 3' wide
A = (9)(3)(2) = 54 ft^2
q = (q/A)A = 6750 B/H

In this concept these are horizontal runners on which panel is deployed. This reduces the weight of the structure to hold up the panels.

FIGURE 14 VSD'S RECOMMENDED DEPLOYMENT APPROACH

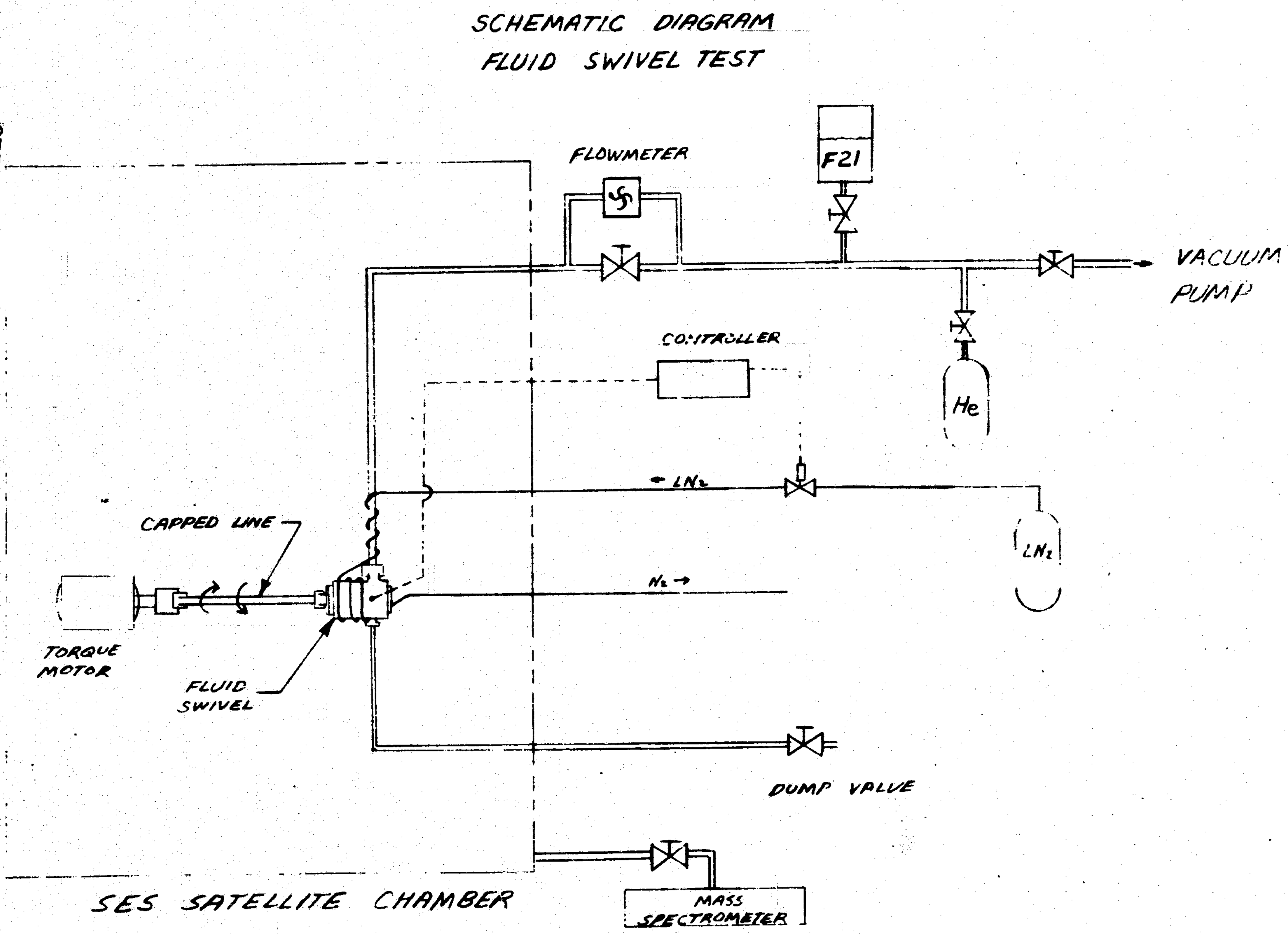


FIGURE 15 SCHEMATIC DIAGRAM - FLUID SWIVEL TEST

photography. If absolute leakage values are required below the range of the liquid flowmeter, a gas rotometer can be installed in the inlet line to monitor helium flow into the swivel joint. Variations in required torque with environmental conditions can be measured by monitoring current to the torque motor. Elevated temperature operation can be verified using the same setup, with or without the chamber, by switching from LN_2 to heated water as the conditioning fluid. For meaningful results, three each of any candidate design should be tested for approximately 100 cycles at both high and low temperature extremes. At the current time, one candidate design has been identified with a second possible design under evaluation. It has not been determined whether or not the scope of testing described above is within the budget constraints of the current program.

2.6 Contact Heat Exchanger

A heat exchanger is required to interface the SCHRM with an experiment module requiring the heat rejection capability of the SCHRM. The heat exchanger could be a fluid-to-fluid type which can be made very compact; however, if the two fluid systems heat exchanger cores are separated,compactness is destroyed and the cores must be brought back together through a contact surface. One obvious advantage of the contact type heat exchanger is the separate sealed systems and the elimination fluid hook-ups. There are disadvantages also and these are associated with the high heat transfer contact surface required. Current program philosophy is to design the SCHRM with a contact heat exchanger.

Work on a contact heat exchanger for the SCHRM has currently been centered on the applicability of the heat exchanger core data found in Figure 4 of Reference [9]. This data has been reproduced in Figure 16 and presents the Fanning friction factor coefficient and the Colburn j factor for a particular heat exchanger core. Figure 17 presents three of the important thermodynamic properties of Freon-12 as a function of temperature. These properties are combined to form the dimensionless parameter, Prandtl number (Pr). Using (Pr) to the 2/3 power in Figure 16, one can see that the Stanton number (N_{st}) can be obtained for the range of Reynold's numbers plotted. The Stanton number is related to the film heat transfer coefficient (h) by the following definition:

$$N_{st} = \frac{h}{G\ Cp}$$

FIGURE 16
PLATE-FIN SURFACE HEAT TRANSFER AND FRICTION CHARACTERISTICS

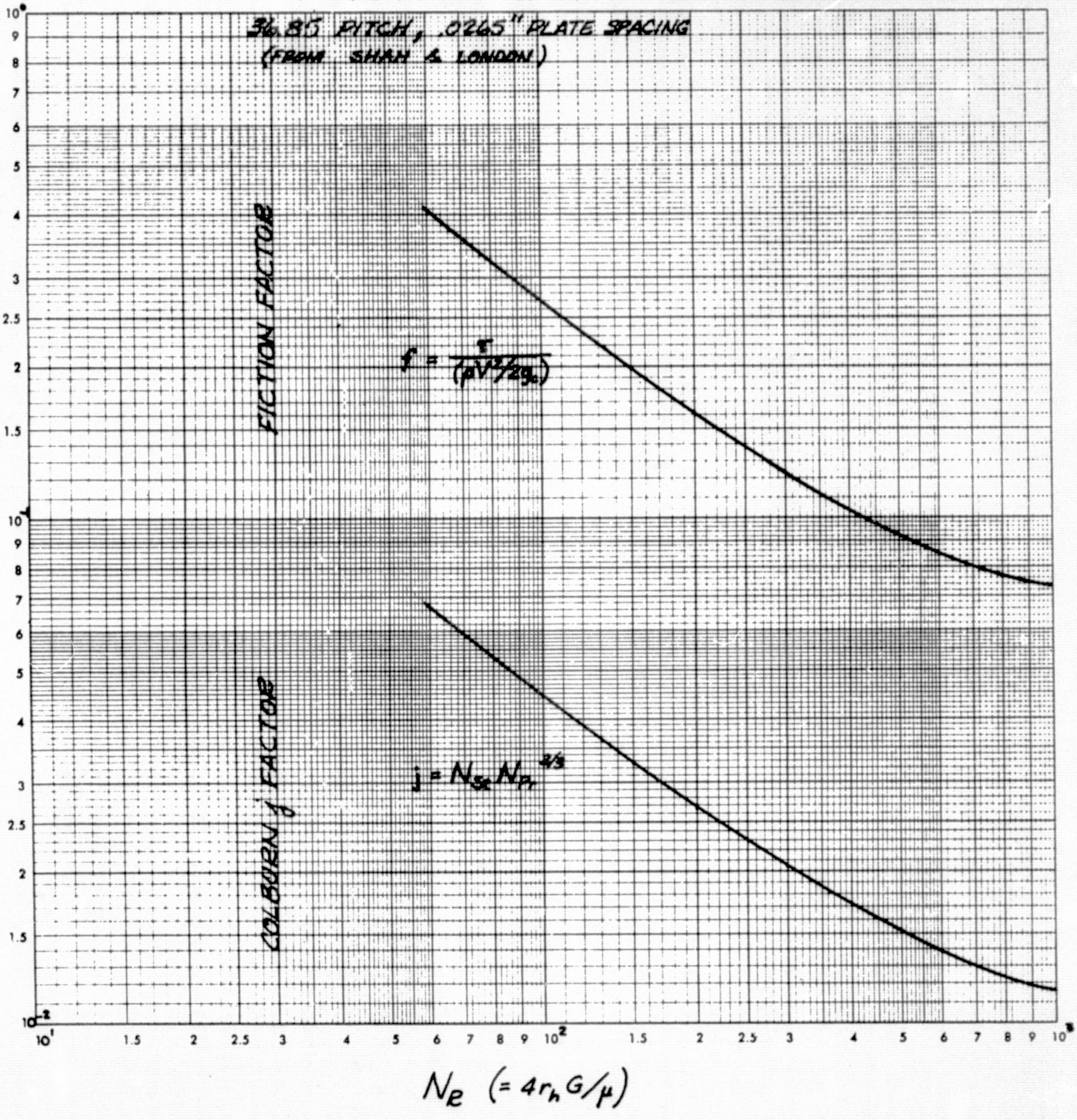
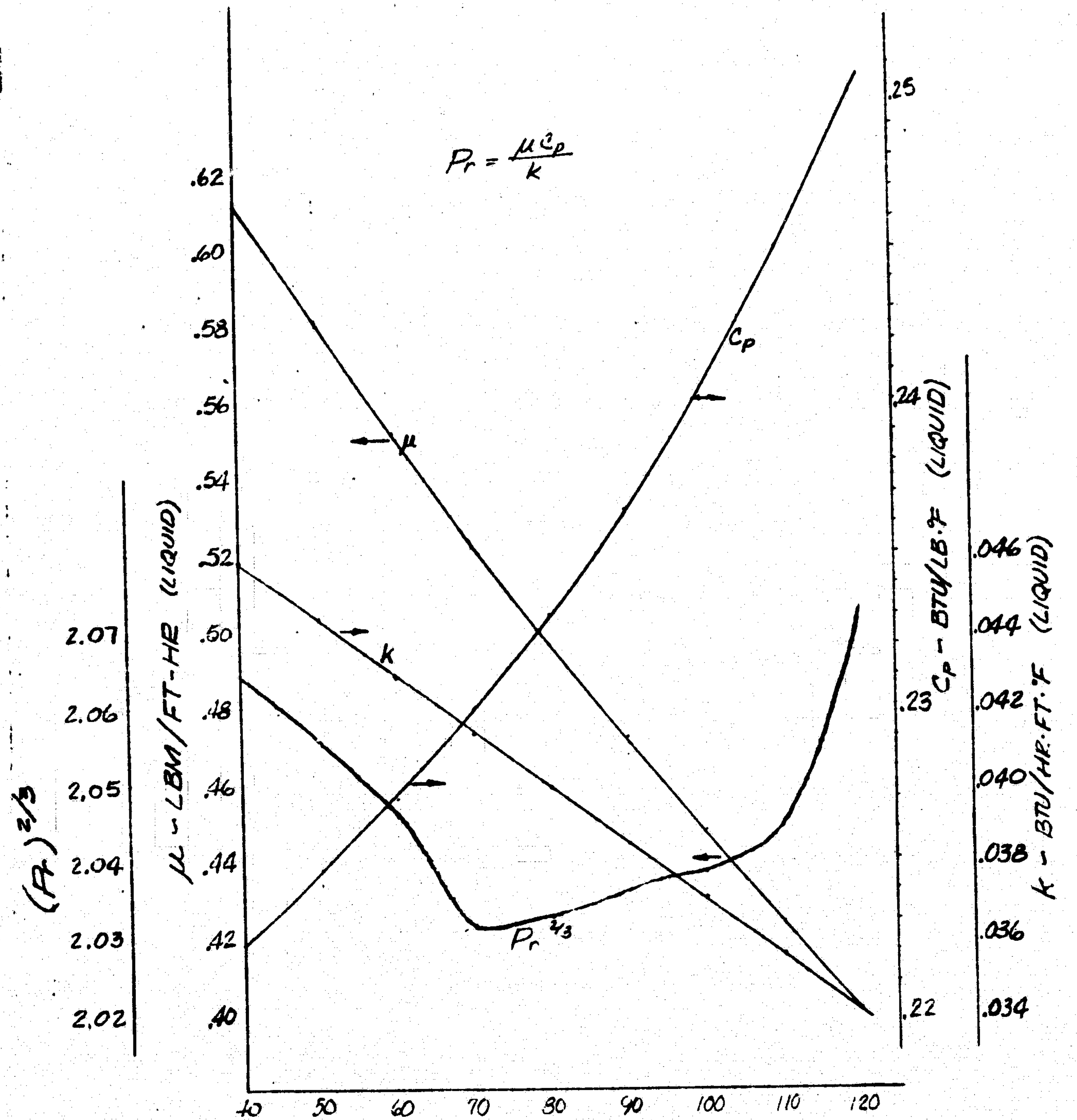


FIGURE 17
THERMODYNAMIC PROPERTIES
OF REFRIGERANT 12



? ? $G = \frac{lb}{ft^2\ hr}$

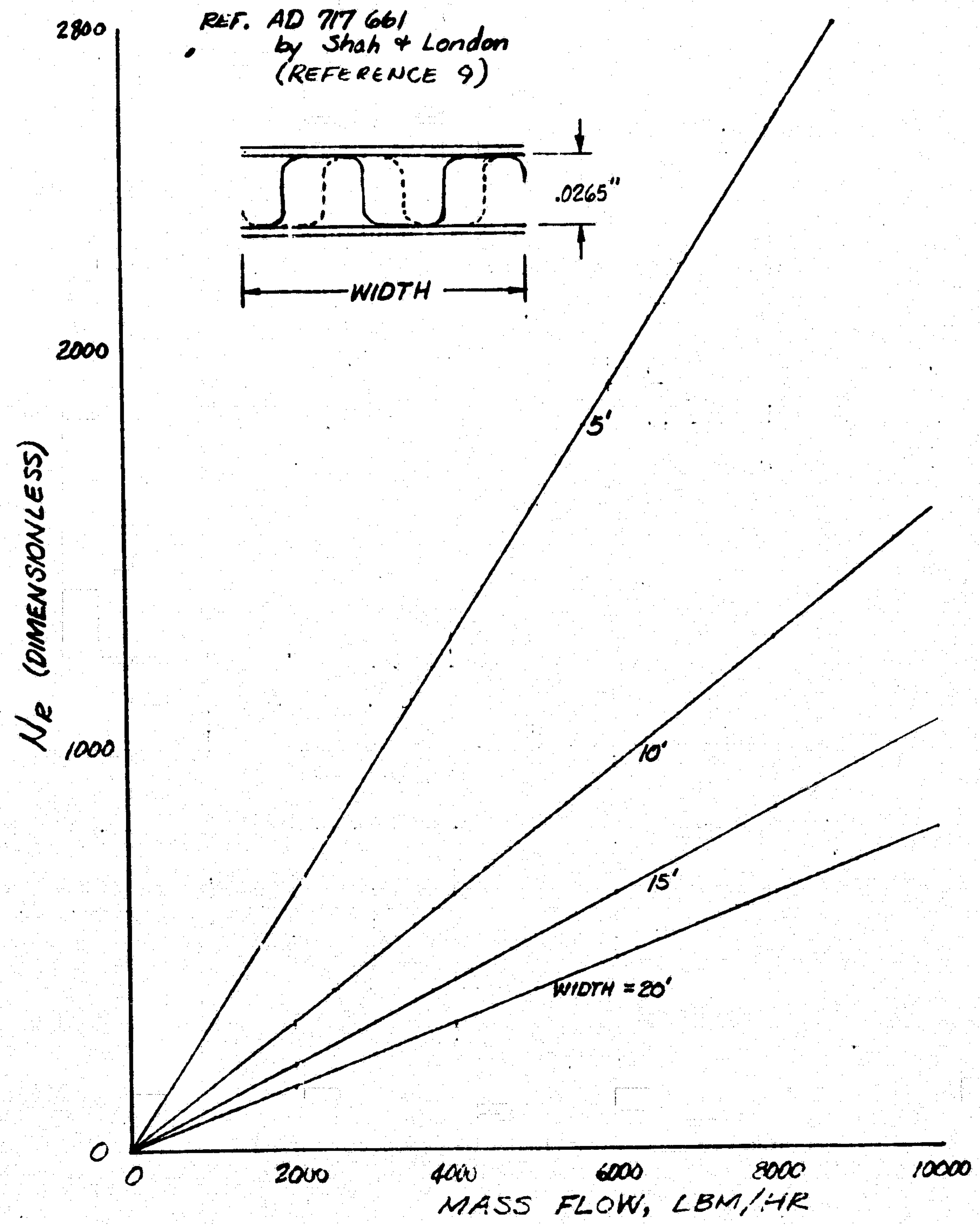
Where: G = volumeteric flowrate, ft^3/hr

Cp = constant pressure specific heat, BTU/lb-°F

As stated previously, the entry into Figure 16 to read j requires a value of Reynold's rumber (N_r). Reynold's number as a function of mass flowrate and core cross-section in plotted in Figure 18. Since the core height is constant, the variation of the cross-section is dependent only on the heat exchanger width. Figure 19 is used to estimate the contact area once the film heat transfer coefficient is established. The hA was assumed to be equal for both sides of the heat exchanger and the contact conductance equal to 500 BTU/hr-ft^2-°F.

Additional analyses are needed to refine the present numbers and calculate pressure drop through the heat exchanger. Analyses in this area will continue into the next reporting period.

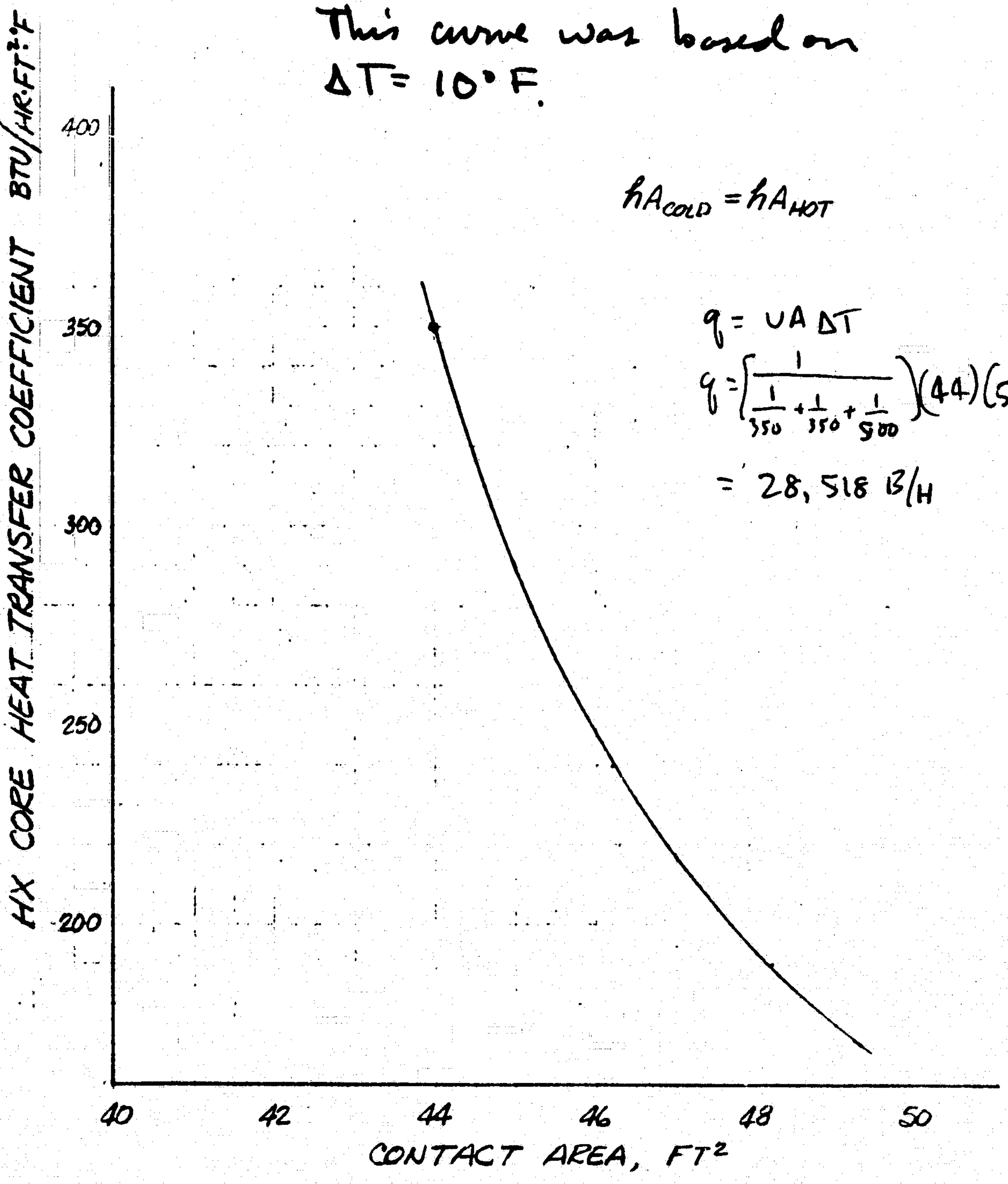
FIGURE 18
N_R FOR 36.85 PITCH PLATE-FIN CORE



what is q for this curve?

FIGURE 19
CONTACT AREA REQUIRED FOR CONTACT CONDUCTANCE OF 500 BTU/HR·FT²·F

This curve was based on $\Delta T = 10°F$.



$hA_{COLD} = hA_{HOT}$

$q = UA\Delta T$

$q = \left(\frac{1}{\frac{1}{350}+\frac{1}{350}+\frac{1}{500}}\right)(44)(5)$

$= 28,518\ B/H$

3.0 CURRENT PROBLEM AREAS

More analysis than was originally planned is being expended in the Design Phase of the study. This is due to a change in deployment concept, more stringent than anticipated documentation requirements, and efforts to produce a dual-mode refrigeration system which is competitive under the study groundrules. Also, two-thirds of the travel budget has been expended to date.

The use of the ATM Solar Array for the radiator deployment mechanism, while desirable in many ways, does raise contractual problems. For example, it will not be possible to test the radiators in the VSD chamber if they are attached to the ATM Solar Cell Array deployment system. In addition, it would be very difficult to design radiators to integrate with that system if modification of that system were not permitted, which is likely to be the case.

For these reasons, VSD recommends that a small SCHRM (not a scale model) be fabricated for feasibility testing, and that the baseline system design be rendered to be compatible with the ATM Solar Array deployment system.

4.0 WORK PLANNED FOR NEXT MONTH

The work planned for next month includes:

(1) Resolve the issue of whether or not the ATM Solar Cell Array Deployment System is available, and whether or not it will be used in the SCHRM baseline design.

(2) Complete the Radiator Panel fluid passage design.

(3) Complete studies on the contact heat exchanger, and make a decision on whether or not it will be used in SCHRM.

(4) Initiate baseline SCHRM mechanical design.

5.0 REFERENCES

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[2] VSD Report No. 00.1599, "Technical Proposal - Development of a Self-Contained Heat Rejection Module", submitted to NASA-JSC, 2 May 1973.

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[5] Carrier, W. H., R. E. Cherne, W. A. Grant, and W. H. Roberts, Modern Air Conditioning, Heating, and Ventilating, Pitman Publishing Company, New York, 3rd Ed., 1959.

[6] Thermodynamic Properties of Refrigerants, ASHRAE, Inc. 345 East 47th Street, New York, N.Y., 1969.

[7] Mason, J. L., W. L. Burriss, and T. J. Connolly, "Vapor Cycle Cooling for Aircraft", WAPC Technical Report 53-338, October 1953.

[8] Keehan, J. H., Thermodynamics, John Wiley and Sons, Inc. New York, 1941.

[9] Shah, R. K. and A. L. London, "The Influence of Brazing on Very Compact Heat Exchanger Surfaces", Stanford Technical Report No. 73, 1 November 1970.

[10] Wolfe, W. L., Handbook of Military Infrared Technology, Office of Naval Research, Dept. of the Navy, Washington, D. C., 1965. pp. 811-824.

[11] Anderson, A. F., et al, "Radiator Design for Space Vehicles" AiResearch Manufacturing Company, Los Angeles, California, circa 1962.

[12] Stevenson, J. A., and J. C. Grafton, "Radiation Heat Transfer Analysis for Space Vehicles", ASD 61-119, 1961.

APPENDIX A

SELF - CONTAINED HEAT REJECTION MODULE

DEPLOYABLE RADIATOR CONCEPT REVIEW

14 SEPT 1973

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION

OBJECTIVES

- EVOLVE DEPLOYABLE RADIATOR CONCEPTS WHICH ARE SUITABLE FOR USE WITH SPACECRAFT NESTED IN THE SHUTTLE CARGO BAY

- SIZE RADIATORS BASED ON STUDY DESIGN REQUIREMENTS

- SELECT THE MOST FAVORABLE CONCEPT FOR DETAIL DESIGN

ENVIRONMENTS

O PANEL COATED WITH SILVER BACKED TEFLON (5 MIL TEFLON)

$\alpha = 0.1$

$\varepsilon = 0.8$

O SHUTTLE OUTER SURFACE RSI

$\alpha/\varepsilon = 0.92$

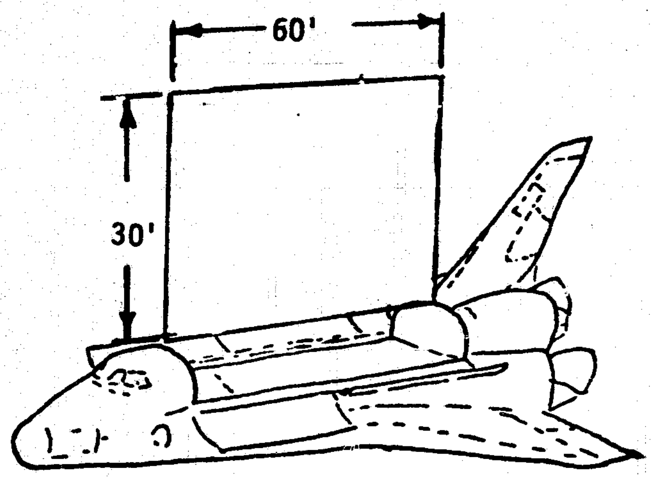
MAXIMUM ABSORBED HEAT FLUX, BTU/HR-FT^2 *

	EARTH ORBIT 100 N.MI.	LUNAR ORBIT 35 N.MI. SUBSOLAR	LUNAR ORBIT 35 N.MI. AVERAGE
FLAT PLATE FACING UP	44.2	44.2	44.2
FLAT PLATE FACING DOWN			
ALBEDO (SOLAR)	14.7	3	3
PLANET EMISSION (IR)	54.8	315	78
TOTAL	69.5	318	81
FLAT PLATE PARALLEL TO SUN'S RAYS			
ALBEDO (SOLAR)	5.5	1	1
PLANET EMISSION (IR)	20.2	116	29
TOTAL	25.7	117	30

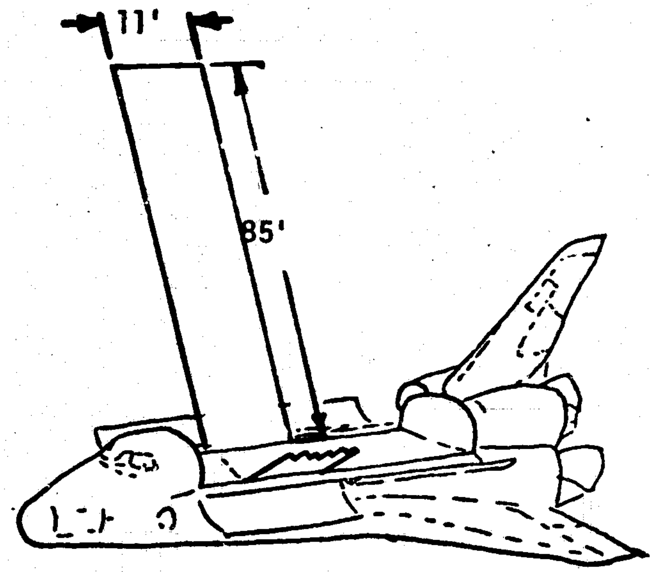
* See References [10] - [12]

ENVIRONMENTS

IR HEAT FLUX FROM SHUTTLE



(a)



(b)

		(a)	(b)
MAX ABSORBED HEAT FLUX ON DEPLOYED RADIATOR,	$\frac{BTU}{HR\text{-}FT^2}$	50	5
REDUCTION IN "VIEW" FROM DOOR RADIATORS, %		18	SMALL

RADIATOR DESIGN ENVIRONMENT

- o $Q_a = 44.2$ BTU/HR-FT^2 (ABSORBED HEAT FLUX)
- o IN EARTH ORBIT THIS IS EXCEEDED ONLY BRIEFLY IN SOME ORBITS
- o IN LUNAR ORBIT IT CAN BE EXCEEDED BY A SIGNIFICANT MARGIN FOR SHORT PERIODS (0.2 HRS.)
- o LUNAR ORBIT WILL REQUIRE SIGNIFICANT ACTION TO DEFEAT ENVIRONMENT
  - USE EXPENDABLE HEAT SINK FOR SHORT PERIODS
  - USE FUSIBLE HEAT SINK FOR SHORT PERIODS
  - ALLOW "PEAKING" AS IS DONE ON APOLLO
  - USE DEPLOYABLE "SHADOW SHIELDS" TO ELIMINATE IR
  - REFRIGERATION IS NOT ATTRACTIVE DUE TO HIGH CONDENSING TEMPERATURE REQUIRED (~250°F)

RADIATOR AREA REQM'TS

APPLICATION	REQUIRED AREA (FT^2)	NO. OF PANELS REQUIRED (TWO SIDE RADIATION) 15ft DIA	10.6x10.6	10.6x20
LOW TEMP (0°F: 80,000 BTU/HR)				
DIRECTIONAL RADIATOR	1566	9*(8.89)	14*(13.9)	8*(7.38)
REFRIG. 140°F COND.	1015	3 (2.88)	5 (4.53)	3 (2.4)
REFRIG. 100°F COND.	1468	5 (4.17)	7 (6.55)	4 (3.46)
MODERATE TEMP (35°F; 200,000 BTU/HR)	3500	10(9.94)	16(15.62)	9 (8.25)
25°F RADIATOR OUTLET	4063			
HIGH TEMP (80°F; 200,000 BTU/HR)	2400	7 (6.81)	11(10.71)	6 (5.66)

* ONE SIDE RADIATION ONLY

SELECTION CRITERIA

O APPLICABILITY

- FRFE FLYING MODULE
- NESTED IN SHUTTLE CARGO BAY
- TEST IN 1-G

O COST

- LOW TECHNICAL RISK
- SIMPLICITY - FEWEST MOVING PARTS
- MANUFACTURING COMPLEXITY
- QUALIFY INDEPENDENT OF PARENT VEHICLE

O FLEXIBILITY

- VARIABLE NO. OF PANELS IN SYSTEM
- WIDE HEAT LOAD RANGE
- VARIABLE PANEL SHAPE
- INSENSITIVE DESIGN
- VARIABLE PLUMBING ARRANGEMENTS
- RETRACTION CAPABILITY

CONCEPT	DEPLOYMENT TECHNIQUES (COULD BE MANIPULATOR ASSISTED)	APPLICABILITY TO NESTED SHUTTLE PAYLOADS	THERMAL PERF. ON SHUTTLE	VOL. UTIL.	POTENTIAL FOR GROWTH IN PANEL SHAPE	TYPICAL DEPLOYED AREAS	HYDRAULIC REQMTS	MANUF. COMPLEXITY	TECHNICAL RISK	COMMENTS
1. Single & Double Pivot, Circular & Cyclic Panel Stacks (Proposal concepts 1 thru 6, pages 65 thru 70)	Hinge W/Motor & Gears and Deployment Arms Where Required	Poor - Panels difficult to deploy unless panel clusters are telescoped out of cargo bay (page 79)	Poor - High radiant interchange with vehicle unless clusters are telescoped out of cargo bay	Good For Free Fliers Limited for shuttle nested payloads	Poor - Considerable redesign of panels & swivel mechanisms if number of panels increased. Limited panel interchangeability.	1617 ft^2 (page 70) to 3500 ft^2	Fluid swivel or coil tubing, multiple pivot points	Very Complex - Hinge joint/swivel machining, circular tube pattern, manifolding	High because of panel shape	Much more attractive on free flyer than on Shuttle
2. Single & Double Foldout (Square, Hexagonal) (Proposal Concepts 7 thru 11, pages 71 thru 75)	Same as 1	Fair - Panel deployment limited unless telescoped out of cargo bay (page 79)	Same as 1	Same as 1	Fair to Good - Swivel mechanism redesign for more panels. Panel interchangeability limited	2538 ft^2 (page 73)	Same as 1	Complex - Hinge joint/swivel machining	Moderate to high	Much more attractive on free flyer than on Shuttle
3. WRAP - AROUND	(a) Arm, Hinge, and Sissors (Like Skylab Wing) (b) Rack & Pinion (c) Sissors (d) Hinge W/Mtr & Gears (e) Astromast Stem	Good N.A. Excellent Excellent	Good in Vert. Deployment Excellent In Horizontal N.A. Excellent (Deployed away from vehicle) Excellent	Fair Good Good	Fair - Curved panel stowage envelopes	3500 ft^2 (both sides, two arrays)	(a) Difficult to use 2-D panel (b) Swivel not required (c) Long flex hose req'd for rack & pinion (d) Control could be achieved by varying the deployed area (e) Swivel not required	Fair to good due to panel curvature Deployment mechanisms may be independent of hydraulic system Utilize existing deployment for Skylab/ATM Solar panels Utilize antenna-solar array deployment technology with Astromast/Stem	(a) Low (b) Low (c) Low (d) Moderate (e) Low	. Deployment technology available . Curvature hard to select for general applicability . Radiator design/qual. relatively independent of vehicle structure/heat loads evolution
4. PLANAR SURFACE STACKS	Same as 3	Good	Good	Fair-depending on payload pallet configuration for nested application	Excellent for rectangular panels with sissor deployment, complete panel interchangeability, easily accommodates aspect ratio and size changes	3500	Same as 3	Good - Flat panels deployment, same as 3	Same as 3	More flexibility on use of deployment mechanisms
5. GULL-WING PANELS	Same as 1	Fair - Clearance problem between space lab type payloads and cargo bay doors as well as potential interference problem with orbiter door/radiator deployment mechanism	Excellent - Radiators deployed outboard of shuttle door	Good-Could be evolved to modular wrap-around approach	Poor - Additional panels reduce diameter of available payload volume	2160 ft^2 (both sides of two panels) (8'x60')	Fluid must cross hinge line and second pivot point	Comparatively Simple - requires similar technology to cargo bay doors/radiator	Low	These type of radiators might best be designed as add-on to orbiter Radiators under cognizance of [illegible] well with independent [illegible] fluid system in cargo bay (i.e., as foldouts from each cargo bay door)
6. ROLL-UP (Like Window Shade)	Stem Astromast Sissors	Good	. Good . Boom req'd to give excellent perf.	Fair-Depends on how tightly the panel can be rolled up	Poor - Roller length and reel size not easily changed	1200 ft^2	. Difficult to use 2-D . Long flex lines req'd . Fluid swivel required	Complex - Roll-up solar array technology may be applicable	High	
7. INFLATABLE	Inflate With Gas	Excellent	. Can be put into excellent location	Excellent	Good - Length dimension can grow without significant redesign	3500 ft^2	. Swivel not required	Very complex - bonding of laminates and tubing difficult	High-presently being developed under NAS9-133[illegible]	. Very Versatile . Mat'ls Problems are Significant . Radiator Perf. Not Demonstrated

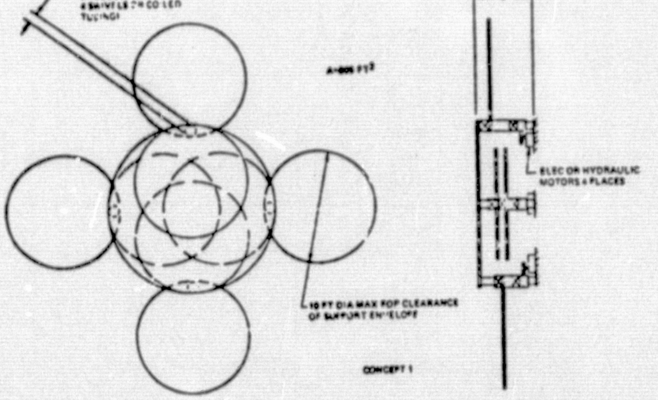


FIGURE 30 FOUR CIRCULAR PANEL DEPLOYMENT CONCEPT

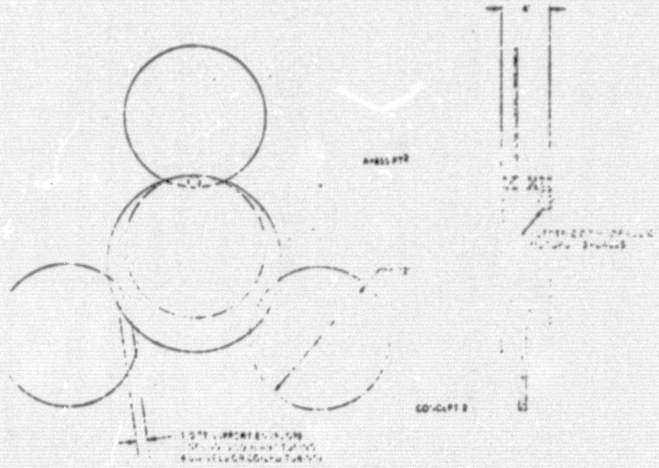


FIGURE 31 THREE CIRCULAR PANEL DEPLOYMENT CONCEPT

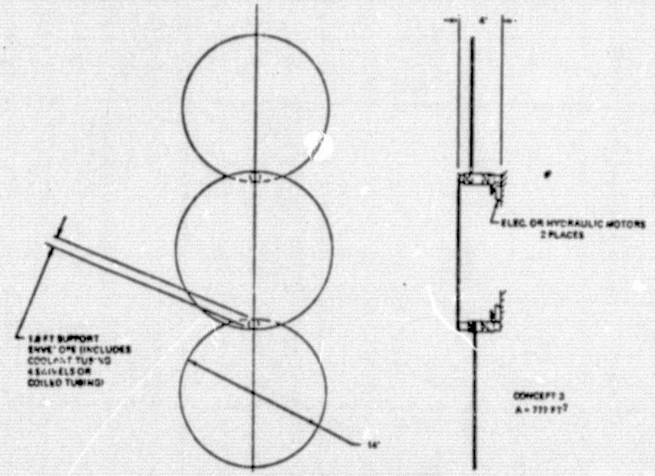


FIGURE 32 TWO PANEL DEPLOYMENT CONCEPT

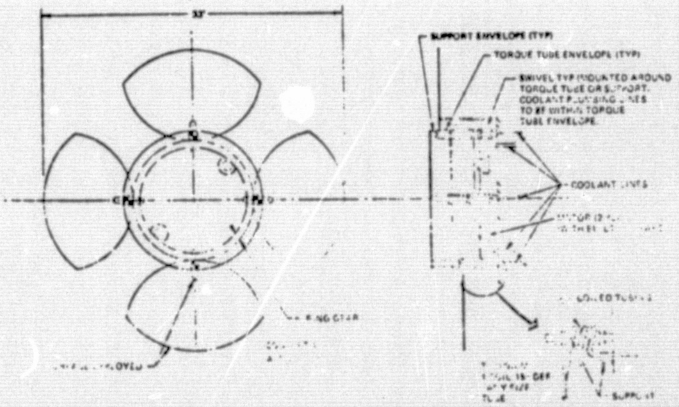


FIGURE 33 FOUR "EYELID" PANEL DEPLOYMENT CONCEPT

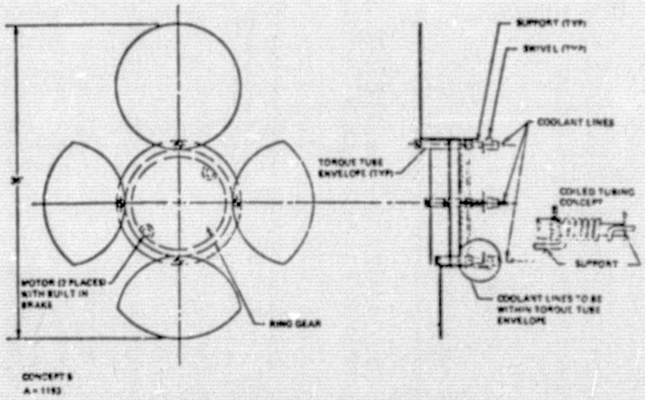


FIGURE 34 THREE "EYELID" PANEL DEPLOYMENT CONCEPT

CONCEPT 1

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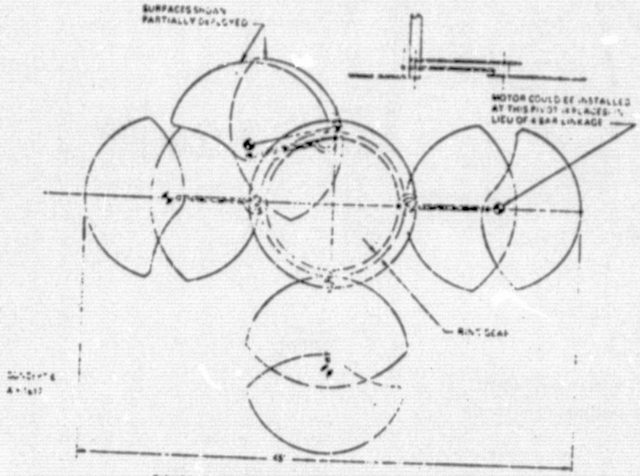


FIGURE 35 DOUBLE DEPLOYMENT CONCEPT EYELID PANELS

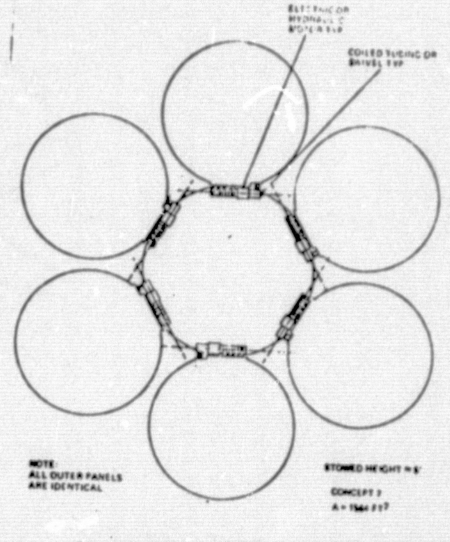


FIGURE 36 CIRCULAR PANEL DEPLOYMENT CONCEPT

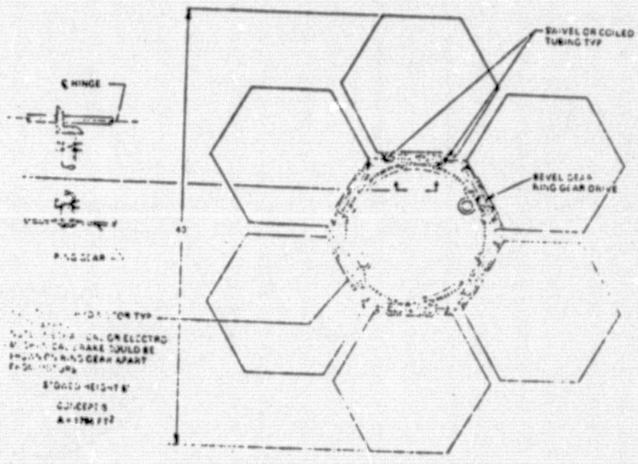


FIGURE 37 HEXAGON PANEL DEPLOYMENT CONCEPT

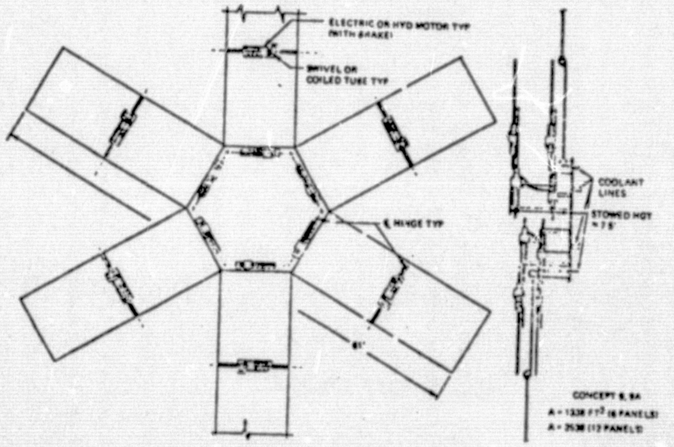


FIGURE 38 RECTANGULAR PANEL DEPLOYMENT CONCEPT

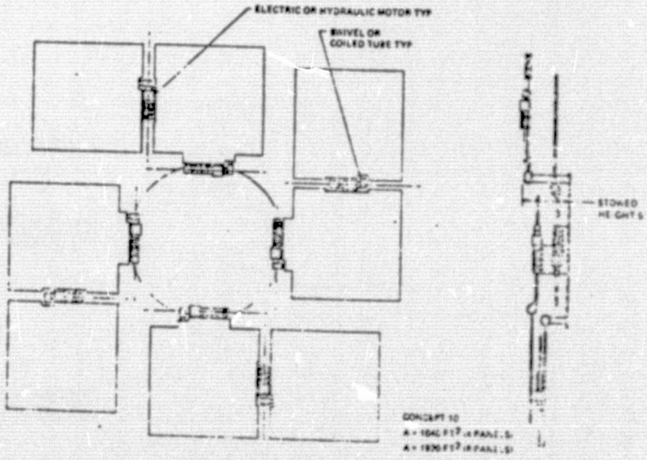


FIGURE 39 SQUARE PANEL DEPLOYMENT CONCEPT

ORIGINAL PAGE IS OF POOR QUALITY

CONCEPT 2

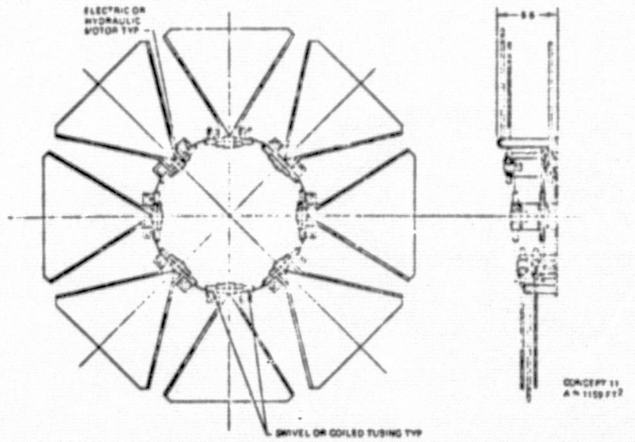
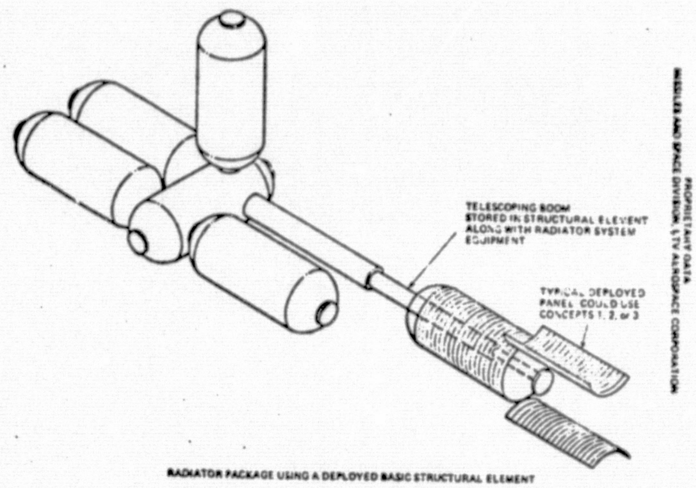


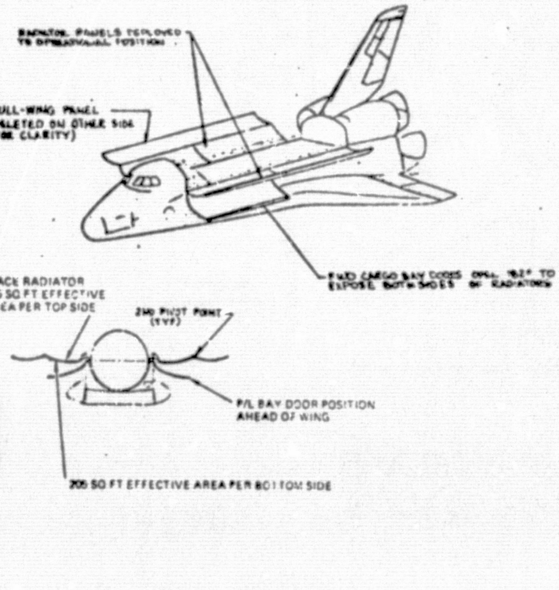
FIGURE 39A TRIANGULAR PANEL DEPLOYMENT CONCEPT

CON(EPT 2



RADIATOR PACKAGE USING A DEPLOYED BASIC STRUCTURAL ELEMENT

COIICEPT 3



CONCEPT 5

SELECTED CONCEPT - #4

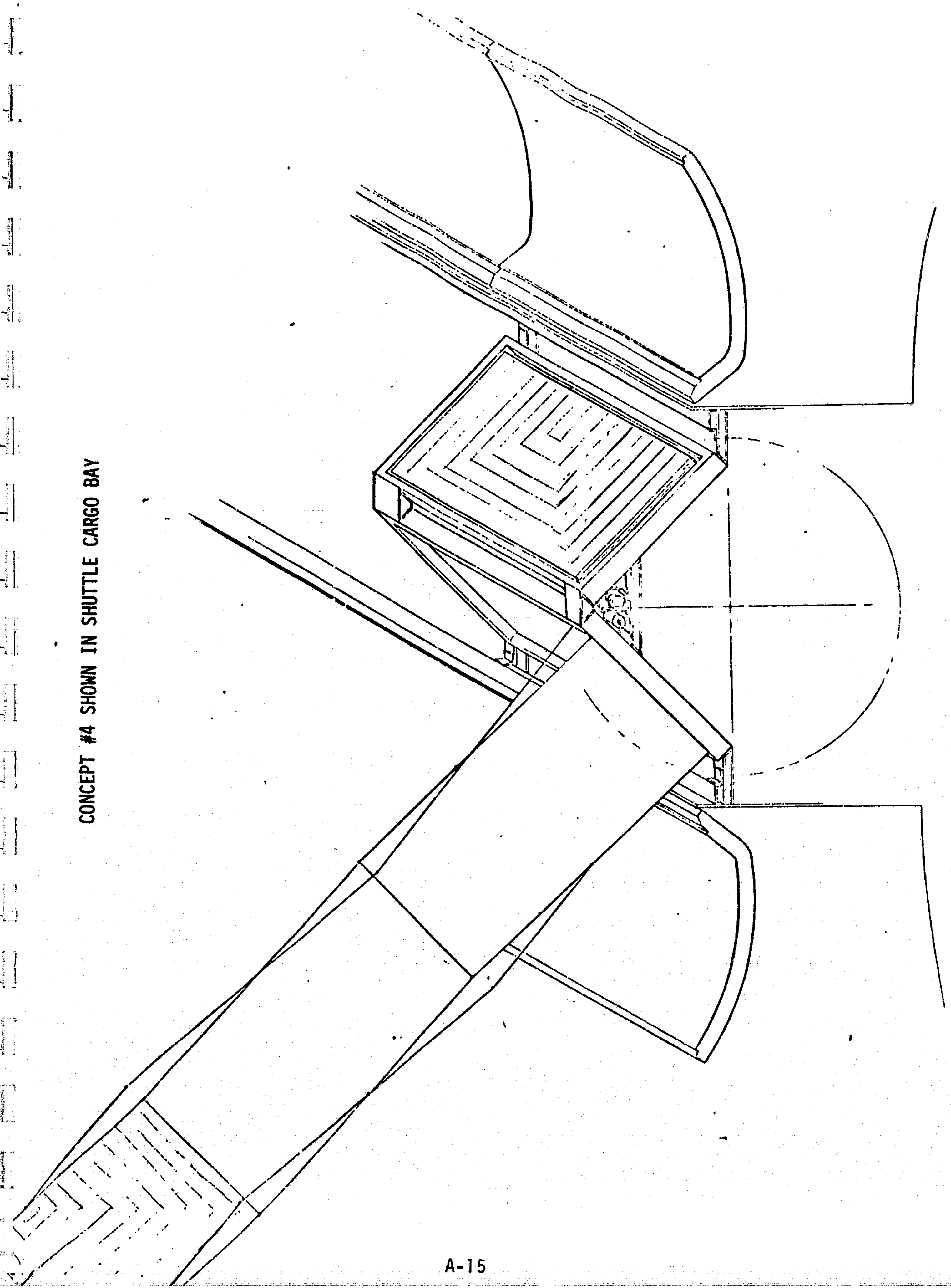
- O FLAT PANELS
- O SCISSOR DEPLOYMENT
- O EXTEND FROM BOTH SIDES OF SHUTTLE

BASIS FOR SELECTION

- O LOW COST
  - SIMPLE MECHANISM
  - FEW MOVING PARTS - MOTORS NOT ON PANELS
  - LOW TECHNICAL RISK
  - LOW MANUFACTURING COMPLEXITY
- O FLEXIBILITY
  - USE IN SHUTTLE CARGO BAY OR ON FREE-FLYER
  - EASY TO CHANGE PANEL DIMENSIONS
  - EASY TO CHANGE NO. OF PANELS
  - CAN USE ONE-D OR TWO-D RADIATOR PANELS
- O PERFORMANCE
  - LOW INTERFERENCE WITH SHUTTLE OPERATIONS
  - GOOD ENVIRONMENT
  - GOOD VOL. UTILIZATION
  - LOW WEIGHT
  - EASY TO RETRACT

NASA INPUTS REQUESTED

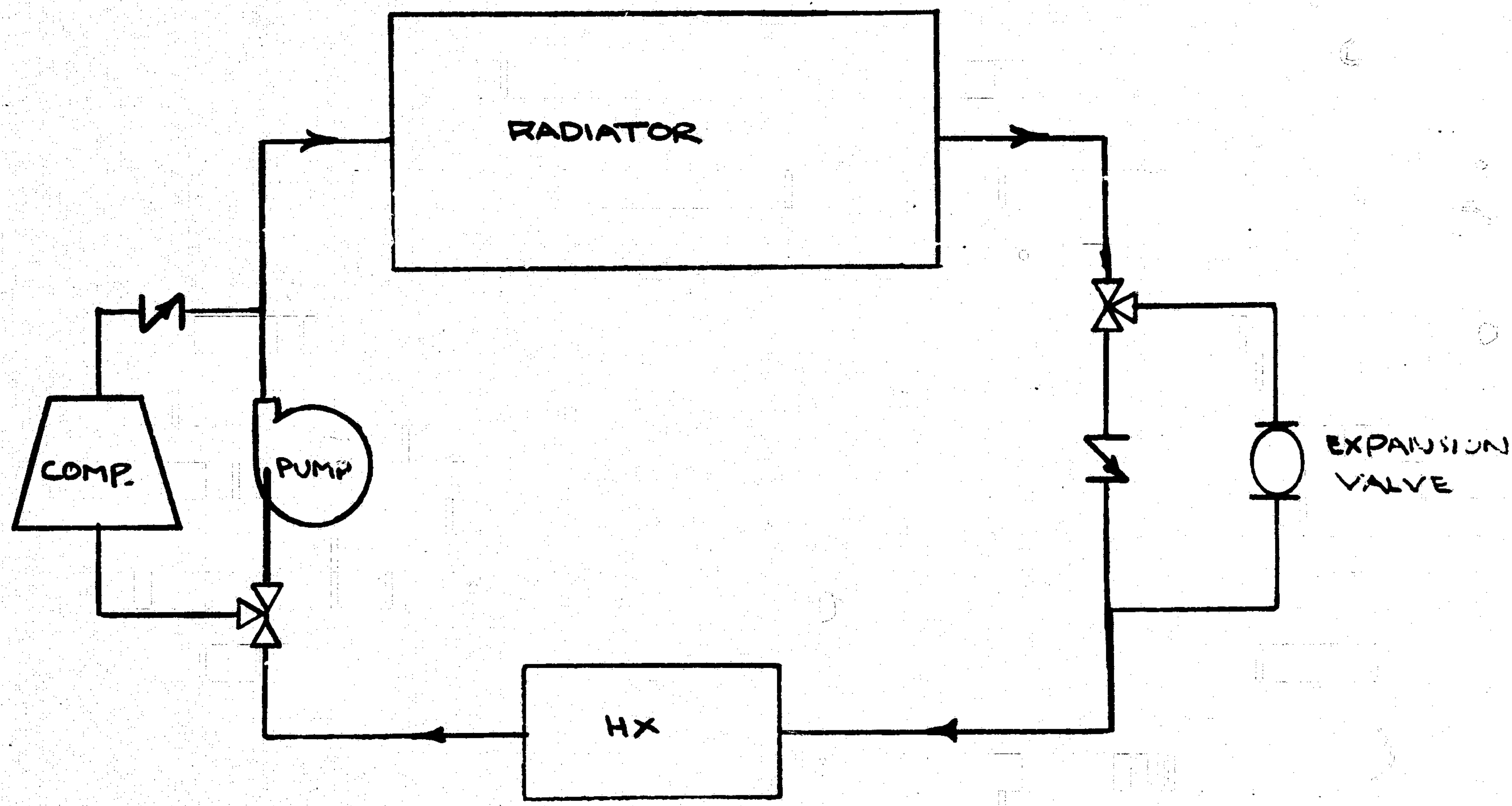
- IS THE SPACE LAB NOW LOCATED AT THE AFT END OF THE CARGO BAY?
- HOW MUCH CARGO BAY LENGTH CAN BE DEVOTED TO THE SHRM? (NOT ALL VOLUME IS USED)
- CAN THE HEAT LOAD ON VEHICLES NESTED IN THE SHUTTLE CARGO BAY BE REDUCED TO 20,000 BTU/HR (WHICH IS MORE CONSISTENT WITH CURRENT PAYLOAD PLANS)?



CONCEPT #4 SHOWN IN SHUTTLE CARGO BAY

FUTURE WORK

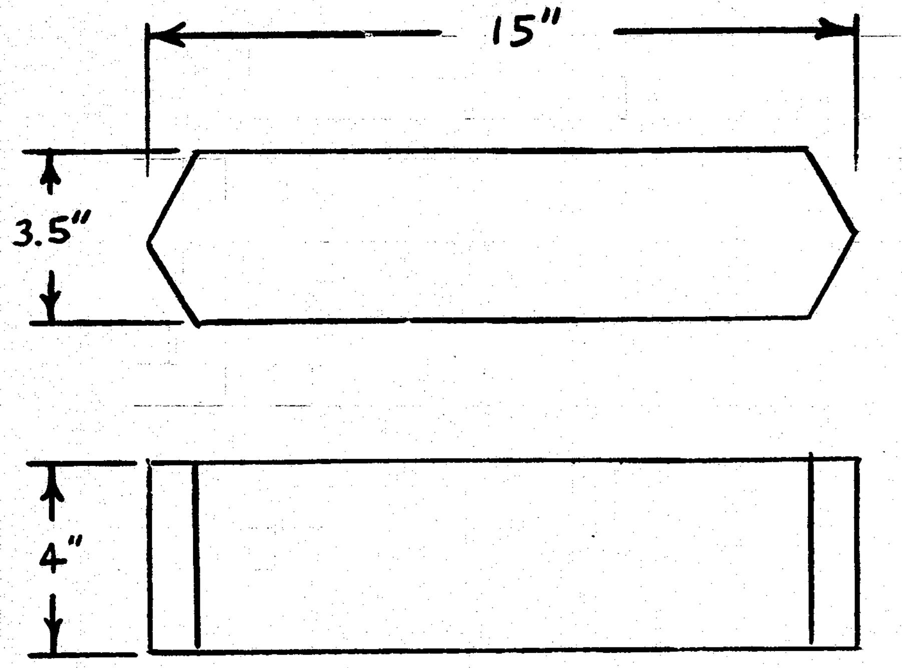
- O DETAIL DESIGN OF RADIATOR DEPLOYMENT MECHANISM
  - SELECT DIMENSIONS
  - SELECT FLEX. JOINT APPROACH (ALL METAL FLEX HOSES WITH 4" BEND RADIUS ARE AVAILABLE)
- O SELECTION OF RADIATOR FLOW ARRANGEMENT
  - TWO-DIMENSION RADIATORS OFFER WIDE HEAT LOAD CAPABILITY
  - TWO-DIMENSIONAL RADIATORS ARE NOT COMPATIBLE WITH CONDENSER OPERATION
  - PANELS COULD HAVE TWO SETS OF FLOW PASSAGES
  - TWO PANEL CONFIGURATIONS COULD BE USED
- O EVALUATION OF CONCEPTS FOR CONTACT HEAT EXCHANGERS
- O SELECTION OF HEAT EXCHANGER
- O EVALUATION OF COMBINED PUMP/COMPRESSOR TECHNIQUES

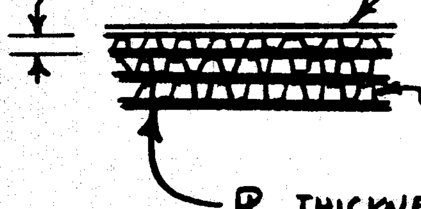


SELF - CONTAINED HEAT REJECTION MODULE

SIMPLIFIED SCHEMATIC

C, 2





FREON 21

$\Delta P \le 2$ psi
$T_{in} = 25°F$
$T_{out} = 150°F$
$\dot{w} = 6250$ pph

WATER

$\Delta P \le 2$ psi
$T_{in} = 160°F$
$T_{out} = 35°F$
$\dot{w} = 1600$ pph

$\varepsilon = 0.923$
NTU = 12.8
Q = 200,000 LB/HR

MAT'L = 347 STLS. STEEL

WT: 12 LB DRY

CONVENTIONAL HX DESIGN

APPENDIX C

DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE

Contract No. NAS9-13533

Report No. T211-RP-003

PROGRESS REPORT

FOR THE PERIOD OF 29 SEPTEMBER through 31 OCTOBER 1973

2 November 1973

Submitted by;

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORP.
DALLAS, TEXAS

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
HOUSTON, TEXAS

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EC/LS Group

TABLE OF CONTENTS

1.0 INTRODUCTION AND SUMMARY

This third progress report on Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 29 September through 31 October 1973.

Effort during this month was primarily concerned with analysis and preliminary design of the heat exchanger equipment in the Self-Contained Heat Rejection Module (SHRM). A trip was taken to Marshall Space Flight Center to review the Apollo Telescope Mount (ATM) solar cell array deployment mechanism. A radiator configuration which is compatible with the ATM array deployment mechanism was evolved.

Work planned for next month includes completion of the preliminary design of SHRM to support the design review on 19 November and initiation of detail design after the review.

2.0 DISUCSSION OF WORK PERFORMED THIS MONTH

2.1 SHRM Flight Prototype Configuration

The SHRM prototype flow schematic is shown in Figure 1. It has two separate flow loops: a high temperature radiator loop; and a dual-mode refrigeration/radiator loop. Each loop carries one-half of the system maximum heat load. The details of the high temperature radiator manifolds and panels are shown in Figure 2, and of the condenser in Figure 3. The high temperature radiator could be of either straight tube or two-dimensional designs. The theoretical performance of these radiators exceeds the assumptions made in the preliminary sizing calculations in that average fin efficiency is over 90%, and the fluid-to-tube temperature drop is around 3°F rather than 10°F. This results in an increase in performance of about 7% over the curves presented in Reference [1]. Heat transfer coefficient as a function of flowrate per tube is shown in Figures 4 and 5 for the high temperature radiator and the radiator/condenser, respectively. Figure 5 shows a comparison of various equations for prediction of heat transfer coefficient in the transition flow regime. The j-factor data were used in this work. For the high temperature radiator with a flowrate of 185 LB/HR-tube, h = 345 BTU/hr-ft^2-°F, and for the radiator condenser with a flowrate of 51 LB/HR-tube, h = 200 BTU/hr-ft^2-°F. Figures 6 thru 9 show that the fluid-to-tube temperature drop is around 3°F or less for the tube spacing of 5-7/8 in. (16 tubes per panel) and the prevailing heat transfer coefficients.

Figure 10 shows radiator panel weight and panel thickness required for a fin effectiveness of 90% (as determined by the method in Reference [11]) as a function of the number of evenly spaced tubes. (The flowrate per tube shown on Figure 10 is one-half the final selected flowrate.) This figure shows that a panel thickness of 0.040 in. provides close to an optimum weight; it was thought that this minimum panel thickness is required to maintain proper stiffness of the panel and ATM deployment mechanism. The flowrate per tube for 16 tubes per panel was found to be satisfactory from a pressure drop standpoint, as discussed later.

During condenser operation the flowrate is 12.5 lb/hr-tube, which is a satisfactory flowrate for an 8-foot tube length according to Reference [12].

Figure 11 (from Reference [12]) shows condensing length vs saturation pressure (or temperature) for various flowrates and inlet gas temperatures. This curve is for an 8-in. wide fin radiating from one side, whereas the current configuration has a 5-7/8 in. fin radiating from both sides. Both fins have over 90% efficiency. Figure 11 is for a case radiating to a 0°K sink; it shows that a tube length of about 8.75-ft. is needed for an inlet temperature of 200°F, a flowrate of 12.5 lb/hr-tube, and a saturation temperature of 120°F. To translate this result to SHRM, it is necessary to account for the variance in fin width, as follows:

$$\text{SHRM} \quad L = (8.75 \text{ ft.})\left[\frac{8\text{-in.}}{(2)(5.875 \text{ in.})}\right] = 5.96 \text{ ft.}$$

The SHRM design is based on an absorbed heat load on the radiator/condenser of 42.8 BTU/ft^2-hr rather than a 0°K sink. Calculations show that a flow length of 94-in. is required for SHRM.

The high temperature radiator and radiator/condenser designs are consistent with the optimum designs given in Reference [13] for a two-sided radiator with a flowrate of 800 lb/panel and 6-in. tube spacing. This is shown by Figure 12, which is taken from Reference [13].

Pressure drop in the high temperature radiator system and in the radiator/condenser system are detailed in Tables 1 and 2, respectively. Typical parametric pressure drop curves used in the design are given in Figure 4 and Figures 13 thru 18. Figure 4 shows the pressure drop across an 8-ft. tube of 3/16" ID for various flowrates. For the design maximum flowrate of 185 lb/hr-tube for the high temperature radiator, the pressure drop is 1.25 psi. Figure 13 shows pressure drop for a 1/8-in. ID tube of 8-ft length as a function of flowrate. For the design radiator/condenser flowrate in the radiator mode of 51 lb/hr-tube the pressure drop is 0.8 psi. Figure 14 (from Reference [12]) shows pressure drop as a function of flowrate for various tube diameters for condensing flow. For the SHRM design with a flowrate of 12.5 lb/hr in a 1/8-in. ID tube, the pressure drop is 0.8 psi for an 8-ft tube length. Figure 15 shows the pressure drop across the swivel joint as a function of swivel throat diameter for R12 vapor, and Figure 16 shows the same for R12 liquid. These curves were generated using techniques and data presented in References [14] thru [17]. Based on these curves a swivel minimum throat diameter of 1/2-in. was selected. Figure 17 and 18 show pressure drop as a

TABLE 1

HIGH TEMPERATURE RADIATOR SYSTEM

PRESSURE DROP SUMMARY

Flowrate = 6000 pph

			Pressure Drop (Psi)
Radiator (Two Wings in Parallel)			
Panels			
16 tubes in parallel, 1.1 psia ea. =	1.1		
2 Headers in series, 0.224 psi =	0.448		
Sub-Total	1.548		
4 Panels in Series		6.2	
8 Swivels in series, 1.26 psi/ea		10.1	
Delivery and Return lines		1	
Contingency		2	
Radiator System Total			19.3
HX			5
Lines and Valves			5.7
TOTAL			30

TABLE 2
DUAL-MODE RADIATOR/REFRIGERATOR
PRESSURE DROP SUMMARY

				Pressure Drop (Psi)
Liquid Phase Operation - Flowrate = 6500 pph				
<u>Radiator</u> (Two Wings in Parallel)				
Panel				
16 tubes in parallel, 0.8 psi		0.8		
Headers;2 in series, 0.1 psi ea.		0.2		
Total per panel		1.0		
4 Panels in parallel			1.0	
1st Swivel			1.72	
2nd Swivel	other swivels are in parallel		0.97	
7th Swivel	with these two; see		0.97	
8th Swivel	Figure 3		1.72	
Lines			1.0	
Contingency			2.0	
Radiator Total				9.4
<u>HX</u>				5.0
<u>Lines and Valves</u>				5.6
TOTAL				20
<u>Vapor Cycle Operation</u> - Flowrate = 1600 pph				
Condenser (4 panels in parallel)				
16 tubes in parallel, 0.8 ea.		0.8		
2 Headers, 0.01 psi ea.		0.02		
Total per panel		0.82		
4 Panels in parallel			0.82	
1st Swivel (Vapor flow),			0.75	
2nd Swivel (Vapor flow),			0.42	
7th Swivel (Liquid flow),			0.05	
8th Swivel (Liquid flow),			0.1	
Vapor line			0.8	
Liquid line			0.2	
Contingency			2.0	
Total Condenser (This compres to an allowable of 10 psi in the vapor cycle design)				5.0
<u>Evaporator</u>				5.0
<u>Valves and Lines</u>				5.0
TOTAL				15

function of tube ID for R12 vapor and R12 liquid, respectively. These curves adjusted for flowrate variation, were used in sizing delivery and return lines, and radiator panel headers.

Based on the pressure drop given in Tables 1 and 2, the pump size for the two SHRM flow loops were established. The specific speeds of the pumps were established, as follows:

$$N_S = \frac{N\sqrt{Q}}{H^{3/4}} \qquad \text{(References [17] - [19])} \qquad (1)$$

N_S = specific speed
N = rotating speed, rpm
Q = flowrate, gpm
H = head, ft-lb_f/lb_m

A rotating speed of 11,400 rpm was selected, since it is very convenient with 200 v, 400 cycle, 3-phase power that is usually available on aircraft and advanced spacecraft such as the space shuttle.

For the high temperature system:

$$Q = \frac{\left(6000\ \frac{LB}{HR}\right)\left(\frac{hr}{60\ min}\right)\left(7.48\ \frac{gal}{ft^3}\right)}{83\ lb/ft^3} = 9\ gpm$$

$$H = \frac{(30\ lb_f/in^2)(144\ \frac{in^2}{ft^2})}{83\ lb_m/ft^3} = 52\ \frac{ft\text{-}LB_f}{LB_m}$$

$$N_S = \frac{(11{,}400)\sqrt{9}}{(52)^{3/4}} = 1766$$

From Figure 19, this corresponds to a pump efficiency of 58%. For this specific speed, there is not much incentive for a two-stage pump, due to the high efficiency of a single stage pump. The pump power requirement is:

$$W = \frac{H\dot{w}}{\eta_P} = \frac{52\ \frac{Ft\text{-}LB_f}{LB_m}\quad 6000\ \frac{LB_m}{hr}}{\left(0.58\right)\left(778\ \frac{ft\text{-}lb}{BTU}\right)\left(3.413\ \frac{BTU/hr}{Watt}\right)} = 202 \qquad (2)$$

It is assumed that a "canned" stator type electric motor will be submerged in the pump. The efficiency for this type of motor is about 75%, and the windage losses are 15 watts (Reference [18]). The total power required by the pump is then:

$$P = \frac{W + Losses}{Efficiency} = \frac{202W + 15W}{0.75} = 290W \qquad (3)$$

For the radiator/condenser system pump,

Q' = 9.75 gpm

H = 34.67 ft-lb_f/lb_m

N_S = 2500

η_P = 60%

W = 141W

and, P = 208W

In the refrigeration mode the flowrate in the radiator/condenser loop is 1600 lb/hr. For compressors, specific speed is

$$N_S = \frac{N\sqrt{Q}}{6270\ H^{3/4}} \qquad \text{(Reference [23])} \qquad (4)$$

where; Q = flowrate in cfm

The SHRM compressor then has a specific speed of 0.0056 with a motor speed of 11,400 rpm. According to Reference [23], this specific speed is in the range of acceptable efficiency for rotary compressors. If the motor rotating speed were reduced, a reciprocating compressor would be satisfactory, however, even if the motor speed were increased to 100,000 rpm, a centrifrigal compressor

would have an efficiency of only 60%. The choice of compressor type is thus between rotary, Heli-Rotor, and reciprocating compressors. The reciprocating compressor would be the most efficient, however, it would seem to have lubrication problems in the zero-g environment. Thus the choice seems to be between the rotary and Heli-Rotor types. The best adiabatic efficiency for this type of compressor with a pressure ratio of 3 to 1 is thought to be about 72%. The motor efficiency for the required size of electric motor was taken as 85% (Reference [20]). The power required to drive the refrigeration system is then 9.4 KW.

Accumulators were sized for the two flow systems based on use of R12 and a temperature variation of the radiator from -250°F to 200°F. The results were:

High Temperature Loop; ΔV = 4400 cu.in.
Radiator/Condenser Loop; ΔV = 5400 cu.in.

Accumulators with working volumes this large should be gas pressurized, based on previous work at VSD.

2.2 Contact Heat Exchanger

Several approaches were considered for the contact heat exchanger, as outlined in Table 3. The table gives the advantages and disadvantages of each approach. In order to select an approach it was necessary to quantize the requirements for the SHRM application. Figure 20 shows the requirements for SHRM, which are for Q = 200,000 BTU/hr with an R12 flowrate of 6250 pph and a water flowrate of 1600 pph. (In the contact heat exchanger analyses, the use of high and low temperature radiators was not considered.) The inlet R12 temperature is 30°F and the outlet temperature is 160°F. The water loop inlet temperature is 165°F, and the outlet temperature is 35°F. The required heat exchanger effectiveness is then (from Reference [7]) :

$$\varepsilon = \frac{T_{hot\ in} - T_{hot\ out}}{T_{hot\ in} - T_{cold\ out}} = \frac{165 - 35}{165 - 30} = 0.9629 \qquad (5)$$

TABLE 3

CONTACT HEAT EXCHANGER CONCEPTS

CONCEPT	ADVANTAGES	DISADVANTAGES
Flat Cold Plates	. Simple . Low technical risk . Easy to manufacture . Convenient if the coldplates can be integrated into structure of two modules which are mated	. Very large area requirement . Very heavy . Very large force required
Irregular Cold Plates Sawtooth Pins	. Reduced surface area requirement over flat coldplates - same area . Improved contact pressure	. Large area requirement . Heavy . More technical risk than flat coldplates . More difficult to fabricate than flat plates
Heat Pipes (NOTE: (1) heat pipes are isothermalization devices. This is not a desirable characteristic in a heat exchanger which requires an $\varepsilon \geq 0.9$. (2) the large area req'd in a contact HX is a result of the low contact conductance between the heat	. Reduces surface area required substantially . Reduces contact force req'd substantially	. High technical risk . Difficult to manufacture . At least 27 separate vapor chambers required for 26 NTU HX . Moderate wt. and vol. requirements

TABLE 3 (CONT'D)

CONCEPT	ADVANTAGES	DISADVANTAGES
Heat Pipes (Cont'd) exchange surfaces. The most dramatic use of a heat pipe would be to form it between the two surfaces, thus producing very high conductance without a large pressure between the two surfaces)		
Stacked Cold Plates	. Reduces surface area since both sides of coldplate are used to transfer heat . Reduces force req'd to provide pressure between plates . Easier to obtain high fluid heat transfer coefficient with acceptable pressure drop . Light . Compact . Technical risk is as low as any other contact HX concept . Simple	. Light compared to other contact HX concepts, but still heavy compared to conventional HX's . May be less convenient to mate than other concepts . May be susceptible to damage due to header design required; both flow loop headers must be at same end of HX for counterflow arrangement, which is required.
Gas Pressurization Between Surfaces	. Can be applied to any of the other concepts except the heat pipe . Reduces force req'd to produce a given contact conductance in a vacuum	. Increases complexity of system

and since for $C_{min}/C_{max} = 1$

$$\varepsilon = \frac{NTU}{1 + NTU} \tag{6}$$

so $NTU = 26$

This would be a rather demanding requirement for a conventional heat exchanger, and would be even more so for a contact heat exchanger. Twice as much heat transfer area is required as would be required if the water to R12 outlet temperature difference were increased from 5 to 10°F. Such an increase would, of course, result in a larger radiator heat transfer area. For this initial evaluation the 5°F ΔT was used.

In order to evaluate the contact surface area between the heat exchanger surfaces, Figure 20 was prepared. It shows surface area requirement as a function of contact conductance for various values of h_c in each fluid loop. The definition of h_c is

$$h_c = h_o \frac{A_o}{A_c} \tag{7}$$

where h_c = heat transfer coefficient of the fluid to the contact surface
h_o = heat transfer coefficient of the fluid to the internal area
A_o = total internal heat transfer area
A_c = contact surface area

Figure 20 shows that contact conductance is more significant to contact surface area requirement than is h_c particularly at low conductance levels. This indicates that an h_c of 1000 BTU/hr-ft^2-°F will provide good results and that primary emphasis should be on the contact conductance. Figures 21 and 22 show h_c for two different types of heat exchanger cores. These Figures show that values of h_c = 1000 are easily obtainable, and Figures 23 and 24, which show pressure drop per unit length for the respective cores, indicate that the pressure drop would not be excessive. For the 46.45 T aluminum core the pressure drop is 1.3 psi/ft for an h_c = 1700 BTU/hr-ft^2-°F, and for the 36.85 R stainless steel core the pressure drop is 3.5 psi/ft for h_c= 1800 BTU/hr-ft^2-°F.

Figure 25 (from Reference [21]) indicates that the contact conductance in a vacuum is proportional to the interface pressure. For U_c of 400 BTU/hr-ft^2-°F a pressure of 550 psi is required. Preliminary analyses indicate that compact HX cores can withstand compressive pressures in the range of 500-600 psi, so this is a reasonable value to use. For the area requirement of 180 ft^2 from Figure 20 for U_c = 500 BTU/hr-ft^2-°F and h_c = 1000 BTU/hr-ft^2-°F, a force of 14.25 x 10^6 LB_f. If U_c = 100 is chosen, then the area required from Figure 20 is 500 ft^2, and from Figure 25, the pressure is 150 psi. This gives a force requirement of 10.8 x 10^6 psi. Thus the total force requirement falls very little with increasing area. One obvious approach is to stack the cores, to allow a smaller total force to generate the required pressure between the water and R12 cores. This also increases the amount of contact area available, since both sides of each plate comes into contact with a HX core of the other fluid. This does cut in half the amount of internal heat transfer area associated with each side, so the internal effective heat transfer coefficient must be doubled to maintain the same value of h_c. This should be possible, since there is a fin effectiveness increase associated with transferring heat from both sides of the core. Figure 26 shows the resulting surface area as a function of number of stacks, and Figure 27 shows the decrease in force requirement to produce 550 psi interface pressure as a function of number of stacks. There are means of increasing the value of U_c, other than increasing contact pressure. These include the use of foils and greases in the interface region; these approaches can be applied if necessary.

For purposes of comparison with a conventional heat exchanger, 60 stacks were selected, and a design was rendered. The weight of the heat exchanger, including steel bolts to provide the required force, was determined as a function of plate thickness. Figure 28 presents the results. It shows that a steel exchanger with copper fins would weigh about 600 lb. if plate thicknesses of 0.030-in. were used. (This is a conventional thickness used in aluminum coldplates.) This compares to a conventional heat exchanger weight of 25 lb. If the plate thickness could be reduced to 0.015-in., the weight of the contact heat exchanger would become 350 lb. Increasing the outlet ΔT in the exchanger to 10°F from 5°F (which reduces the NTU requirement

from 26 to 13) would further reduce the weight to around 175 lb.

This contact heat exchanger design has heat exchanger cores which are held together by the manifolds. The two fluid circuits are joined by inserting one set of cores into the gaps in the other set of cores. Bolts are then inserted through holes in the cores, and are tightened to produce the necessary pressure in the interface.

Figure 29 shows a representative design of a SHRM contact HX using very compact HX core material, the performance of which is shown in Figures 22 and 24. The assumed plate thickness for this design is 0.015-in. The pressure drop for the HX is 4.6 psi.

This contact HX concept has good promise for success, however, the weight penalty is substantial. The minimum plate thickness which will provide the necessary conductance must be established. This is the primary factor in the contact HX weight.

2.3 Laboratory Prototype SHRM Configuration

As discussed in Reference [1], a small laboratory unit is required for testing in the VSD space environment simulator (SES). The selected approach is to use panels which are 3-ft x 3-ft, and to use horizontal deployment in the SES. The deployment mechanism and radiator/condenser manifolds are shown in Figure 30. A single wing will be used to demonstrate both the dual-mode radiator/condenser and the high temperature radiator operation.

Heat rejection will be limited to 5555 BTU/hr in either the radiator or refrigeration modes. This corresponds to a flowrate of 267 lb/hr for the high temperature radiator, and 88 lb/hr for vapor cycle operation. At these flowrates it will not be feasible to use prototype pumps and compressors for the laboratory prototype radiator.

3.0 CURRENT PROBLEM AREAS

The program is on schedule and there are no technical problems. A significant decision will have to be made at the preliminary design review with regard to program direction; whether to continue with the plan to build and test the laboratory prototype SHRM, or to bypass the laboratory prototype and proceed to design and test the full-scale prototype SHRM.

4.0 WORK PLANNED FOR NEXT MONTH

The work planned for next month includes:

(1) Completion of preliminary design of the laboratory and flight prototype SHRM's.

(2) Preliminary Design Review on 19 November 1973.

(3) Initiation of detail design of the selected SHRM concept.

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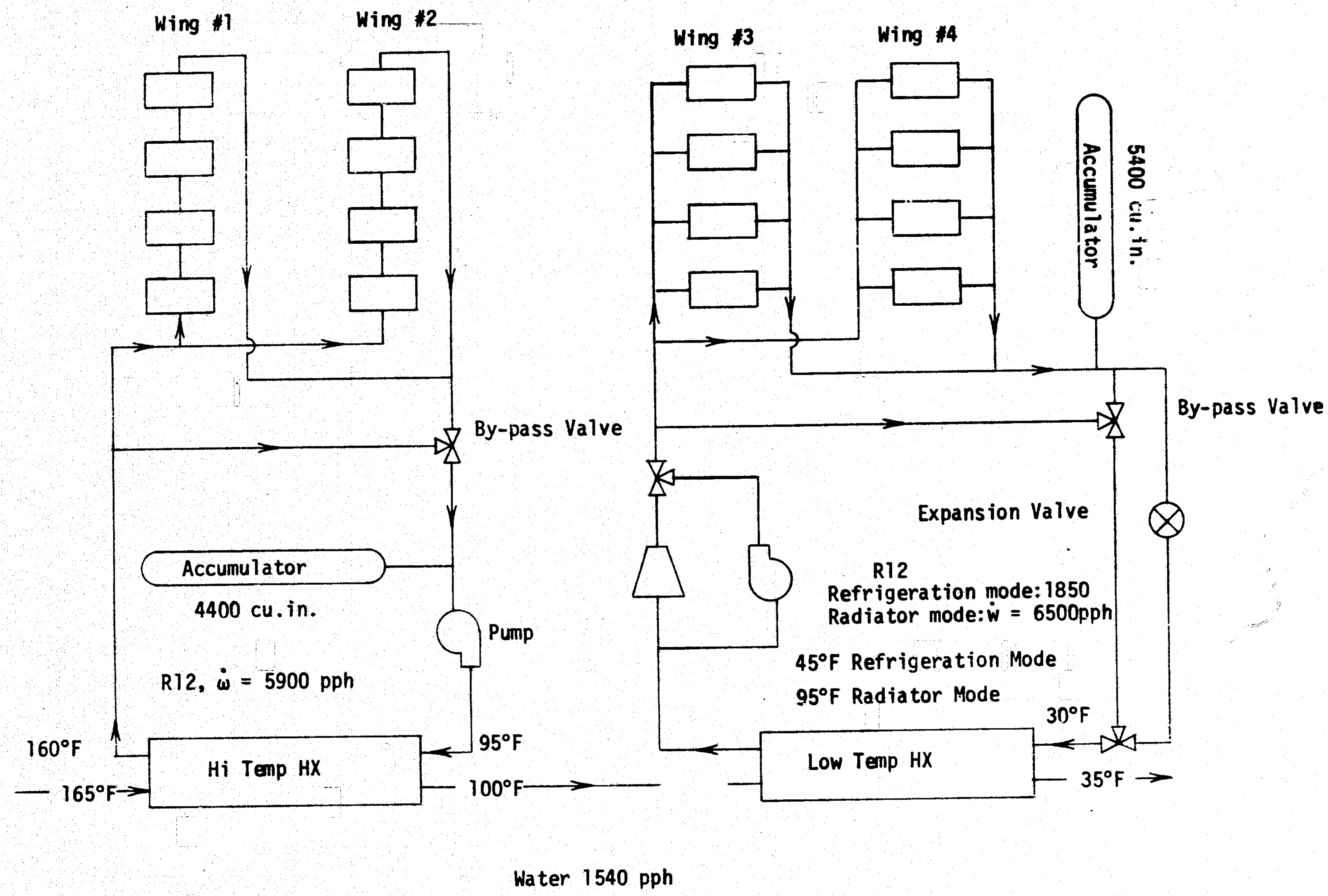


FIGURE 1 PROTOTYPE SHRM FLOW SCHEMATIC

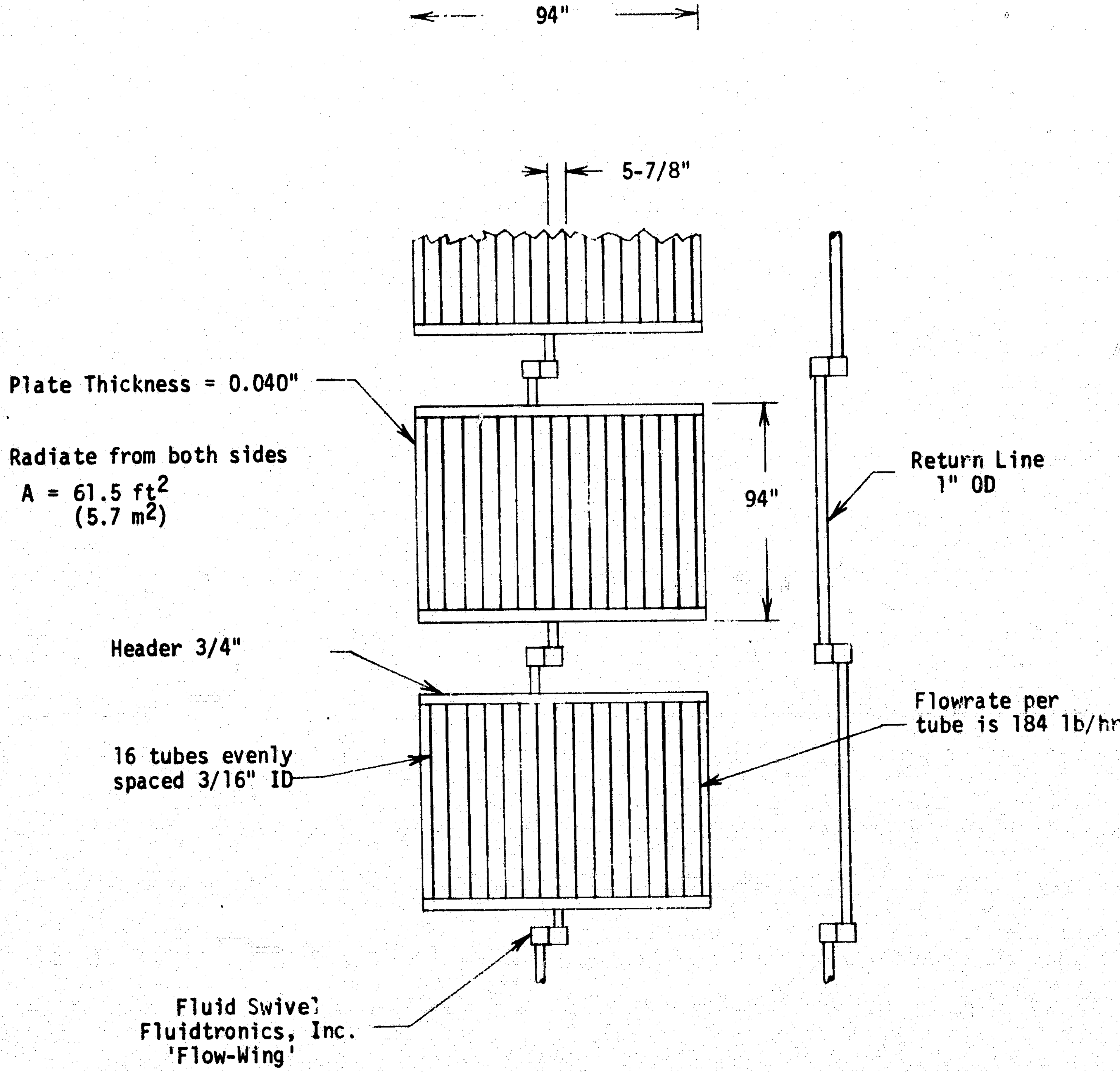


FIGURE 2 HIGH TEMPERATURE RADIATOR TUBE AND MANIFOLD LAYOUT

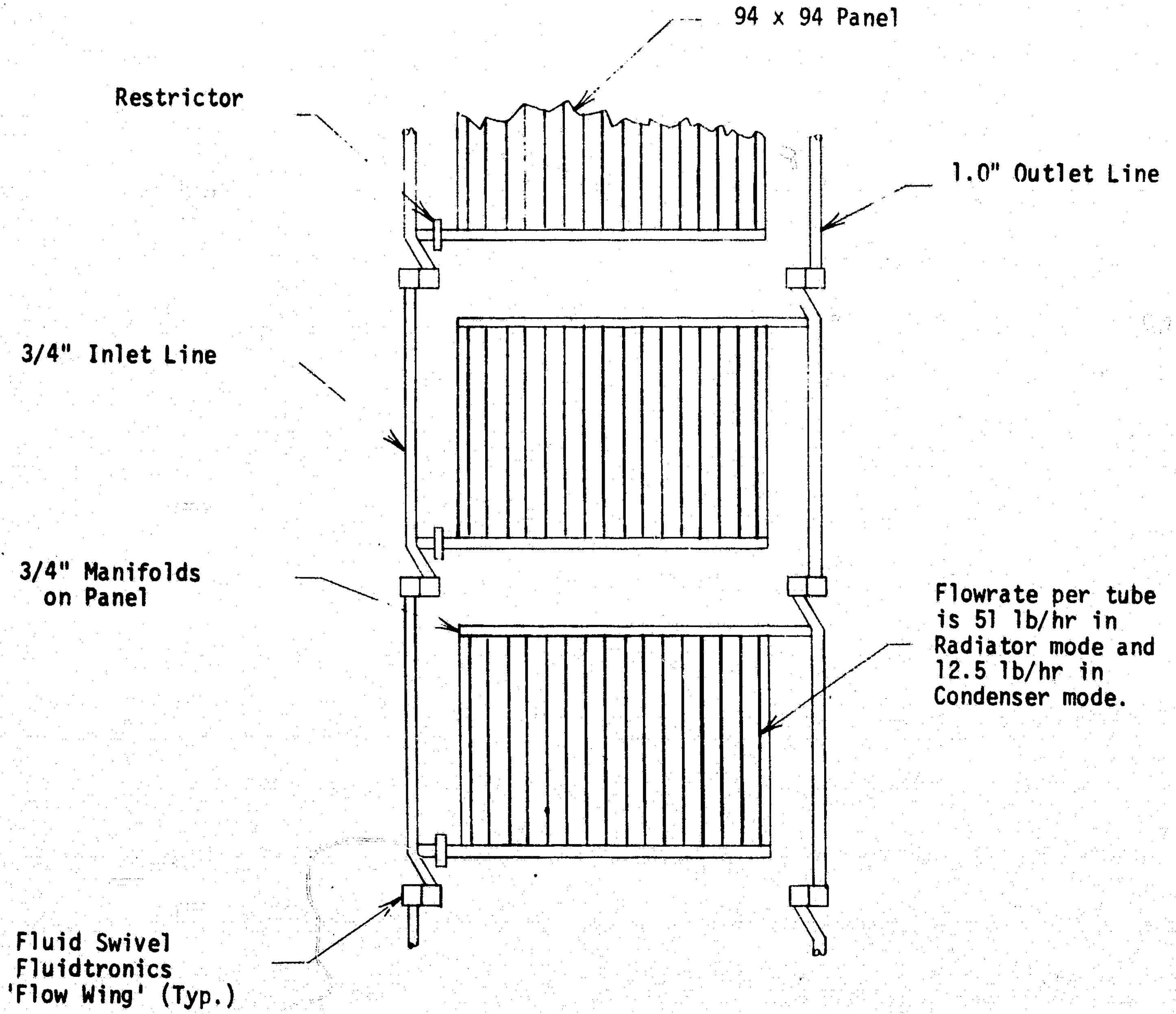


FIGURE 3 RADIATOR/CONDENSER TUBE AND MANIFOLD LAYOUT

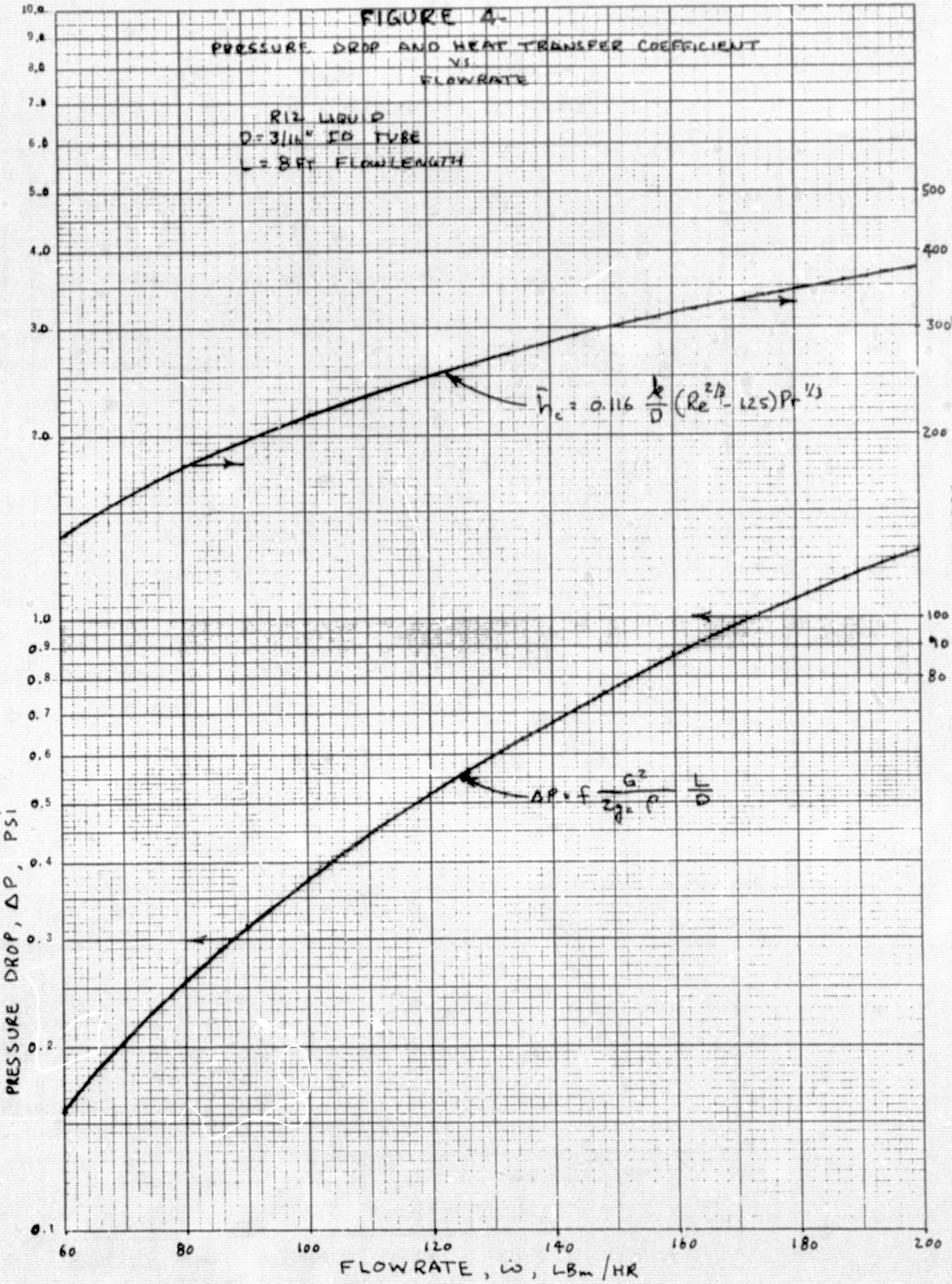
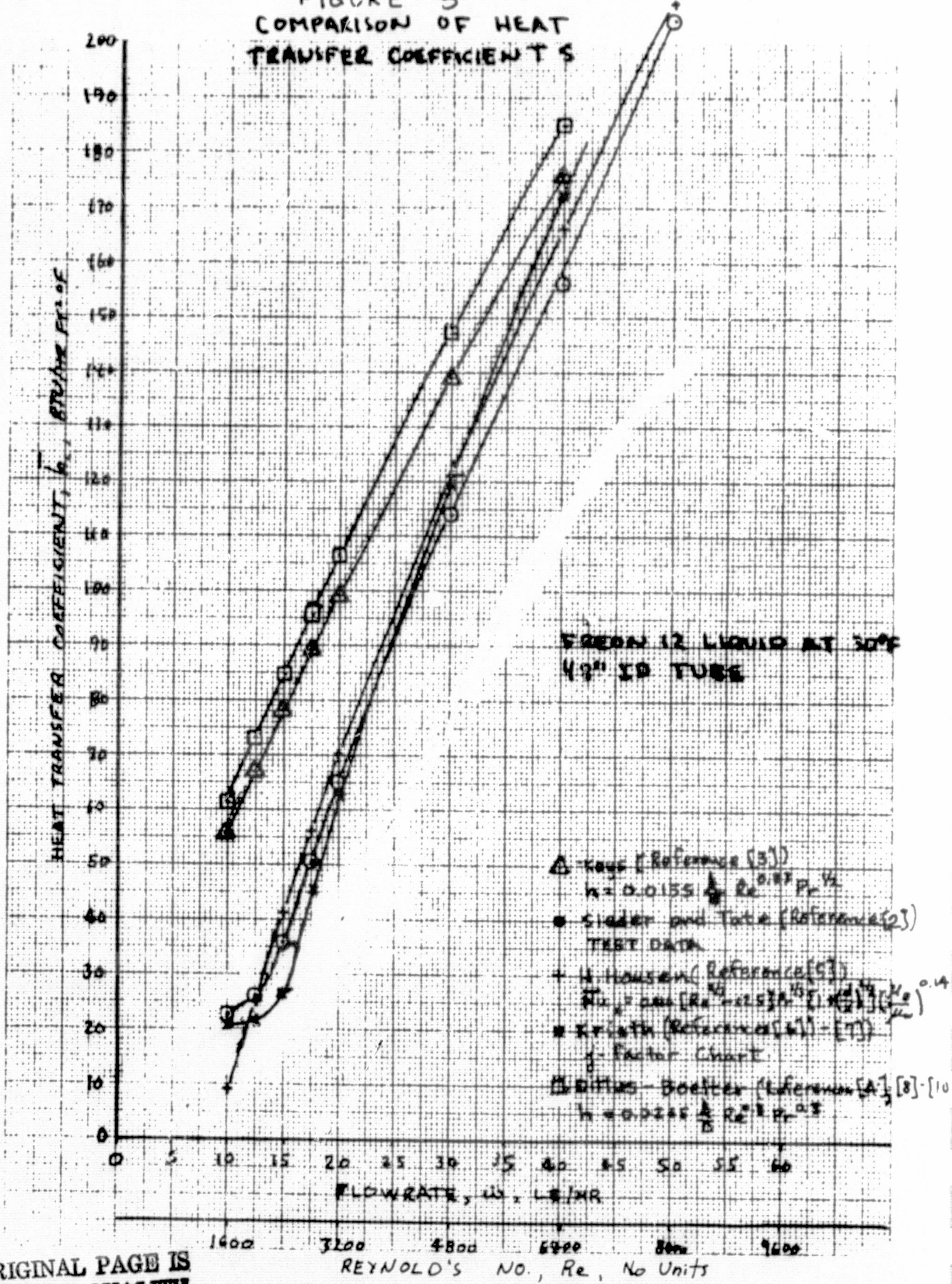


FIGURE 4
PRESSURE DROP AND HEAT TRANSFER COEFFICIENT
VS
FLOWRATE
R12 LIQUID
D = 3/16" ID TUBE
L = 8 FT FLOWLENGTH
$\bar{h}_c = 0.116 \frac{k}{D} (Re^{2/3} - 125) Pr^{1/3}$
$\Delta P = f \frac{G^2}{2 g_c \rho} \frac{L}{D}$
PRESSURE DROP, ΔP, PSI
HEAT TRANSFER COEFFICIENT, h_c, BTU/HR FT² °F
FLOWRATE, ω, LBm/HR
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
500
400
300
200
100
90
80
60
80
100
120
140
160
180
200

FIGURE 5
COMPARISON OF HEAT
TRANSFER COEFFICIENTS



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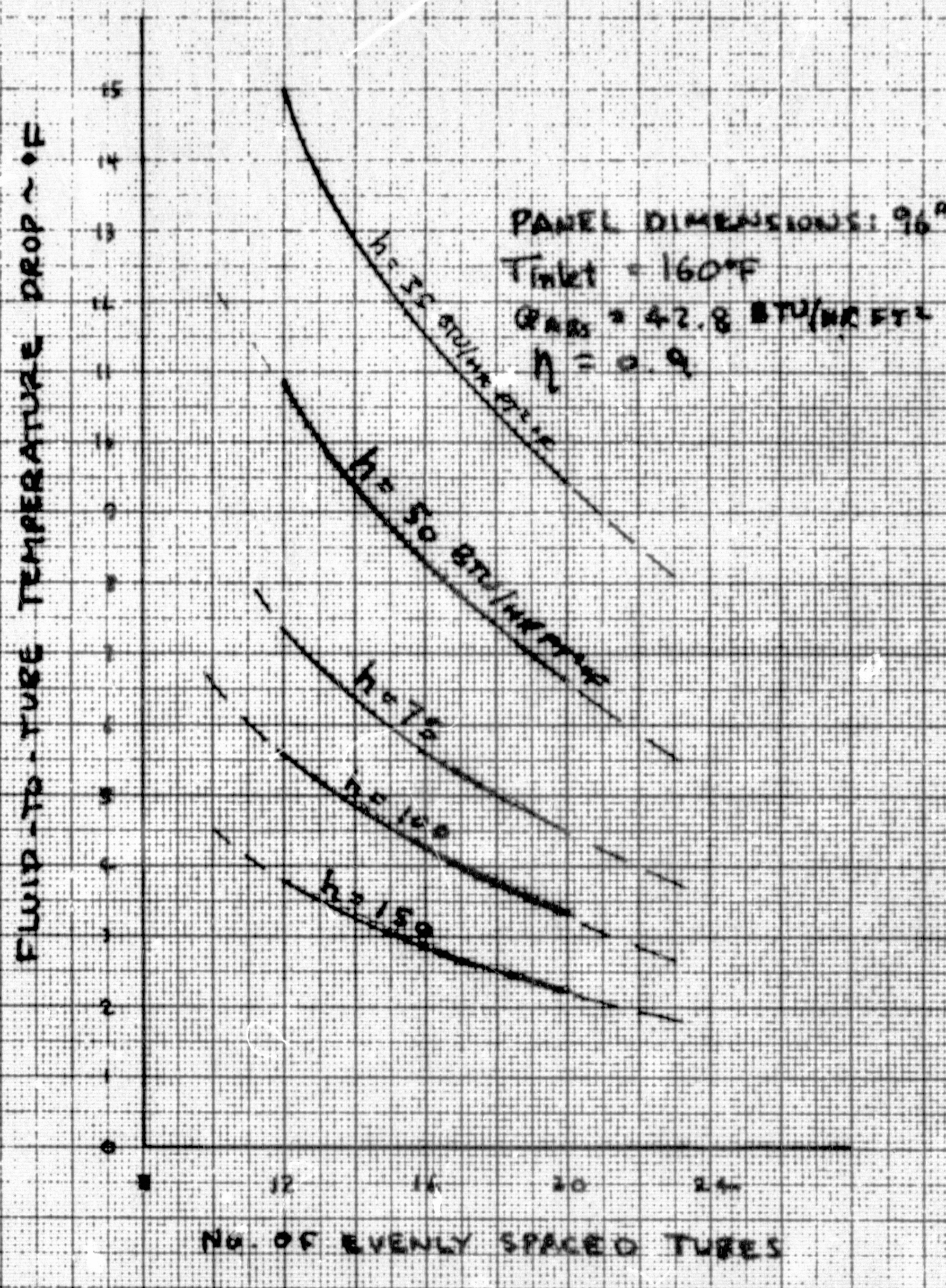

FIGURE 6 FLUID-TO-TUBE TEMPERATURE DROP VS. NO. OF TUBES

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FIGURE 7 FLUID-TO-TUBE ΔT vs $\bar{h}_c$ AT PANEL INLET UNDER DESIGN CONDITIONS

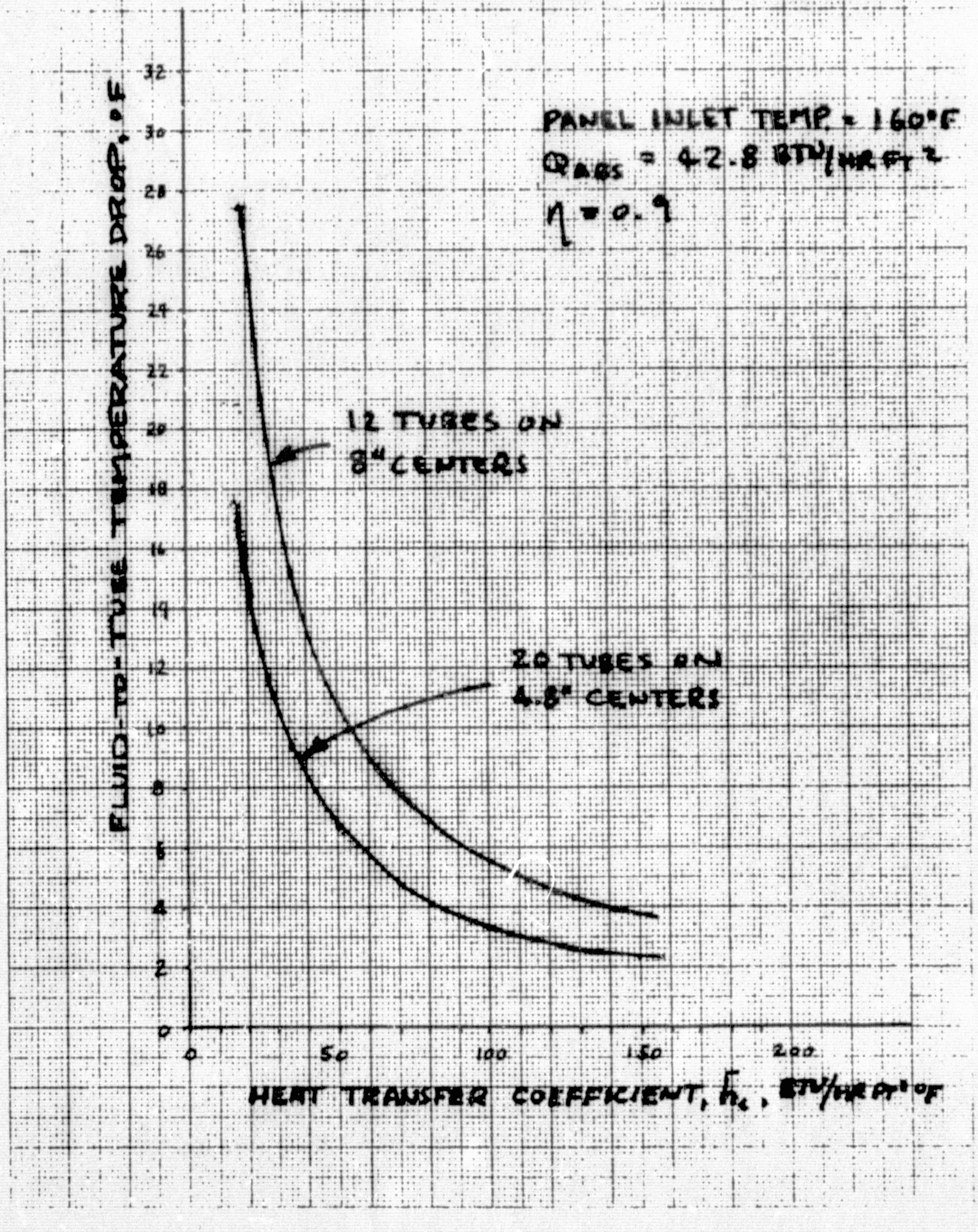


FIGURE 8 FLUID-TO-TUBE ΔT vs. h_c AT PANEL INLET WITH MAX HEAT LOAD AND 0°R SINK CONDITIONS

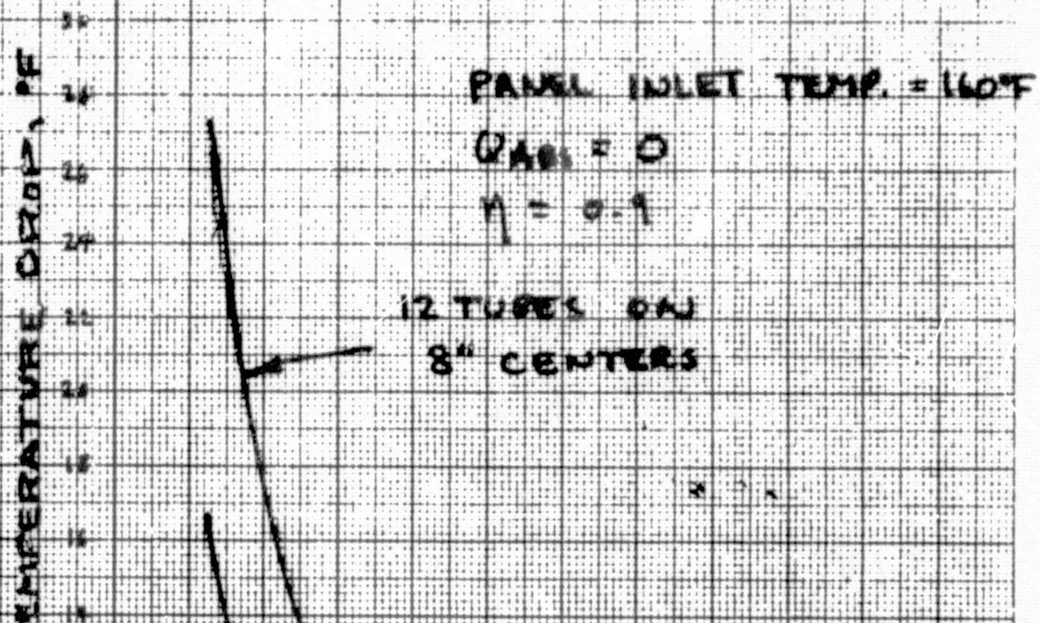
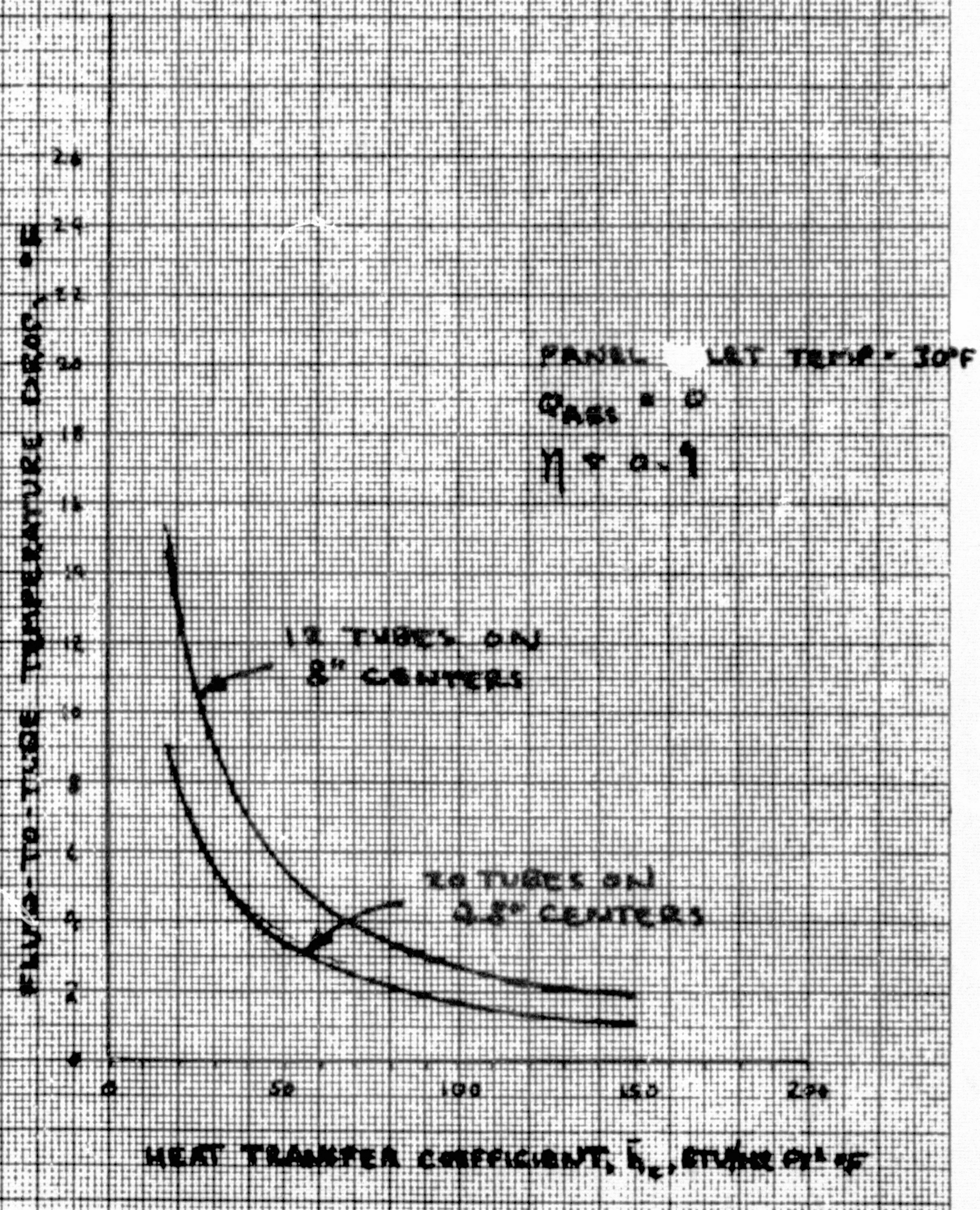


FIGURE 9 FLUID-TO-TUBE ΔT VS. $\bar{h}_c$ AT PANEL OUTLET UNDER MAX HEAT LOAD AND 0°R SINK CONDITIONS



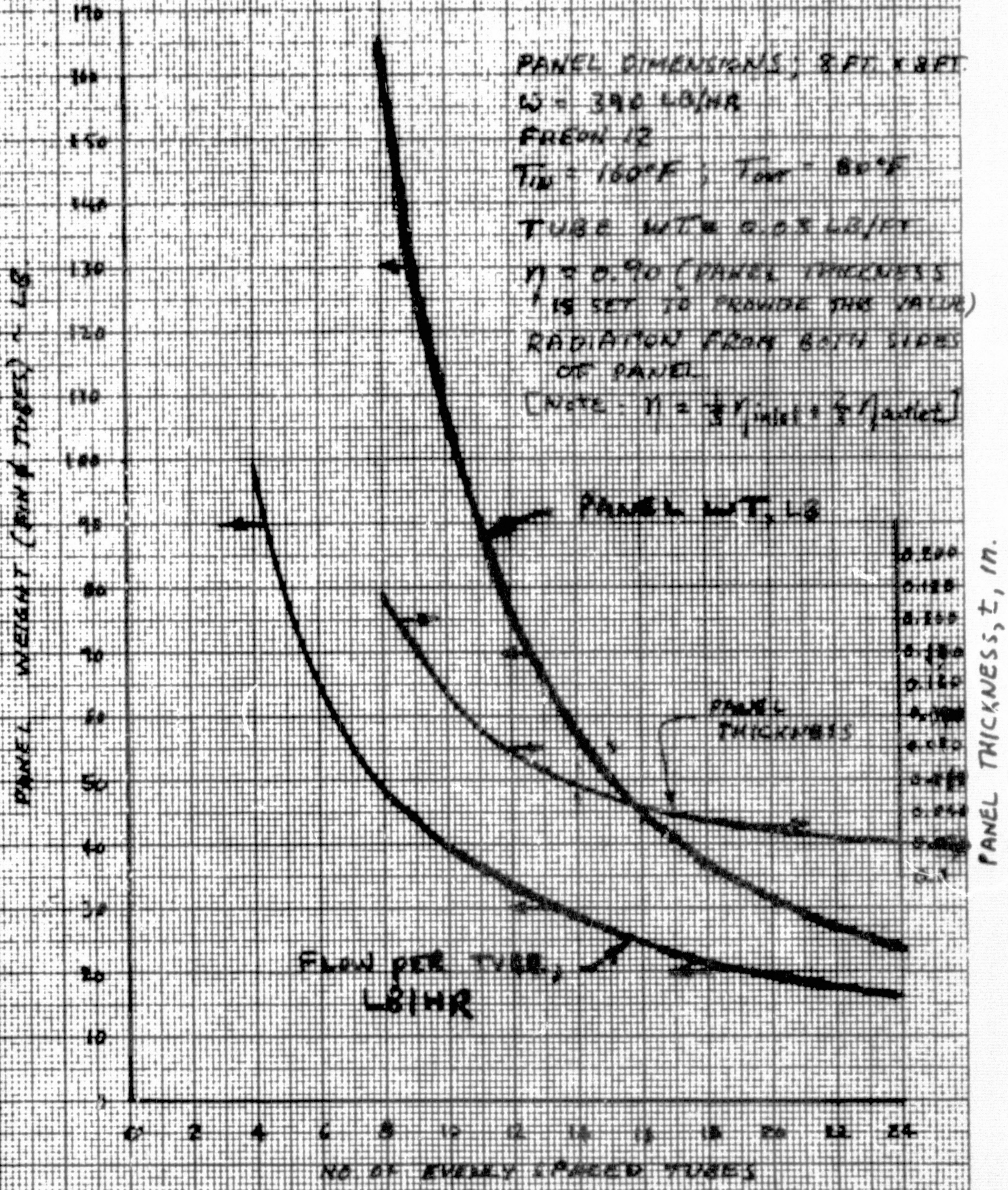


FIGURE 1? PANEL WT. VS NO. OF TUBES

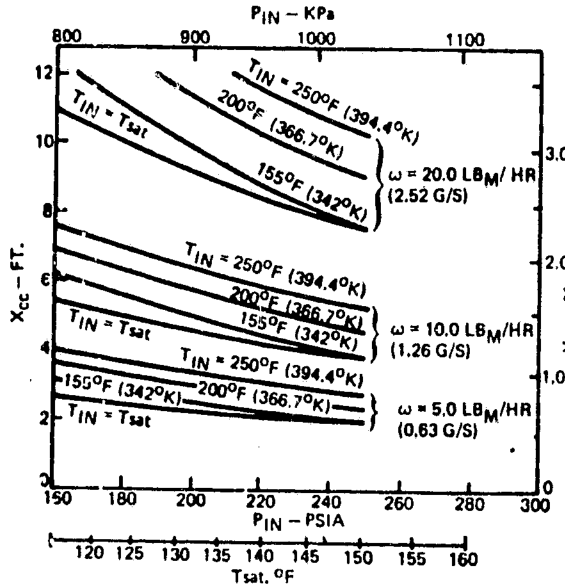


Fig. 11 Condensing length required for various inlet temperature and pressure conditions

NOTE: This curve is for an 8-inch fin, 90% efficient, radiating to a 0°F sink - from Reference [12]

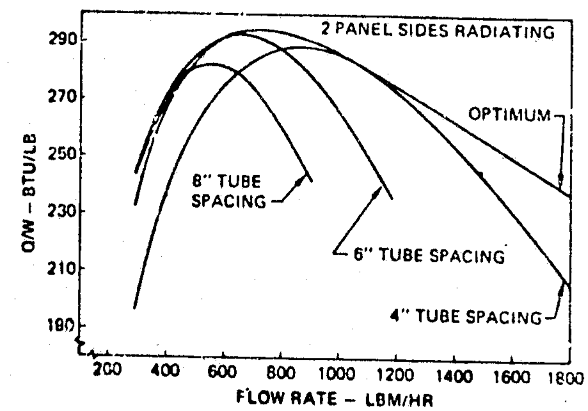


FIGURE 12 OPTIMUM RADIATOR DESIGN

(FROM REFERENCE [13])

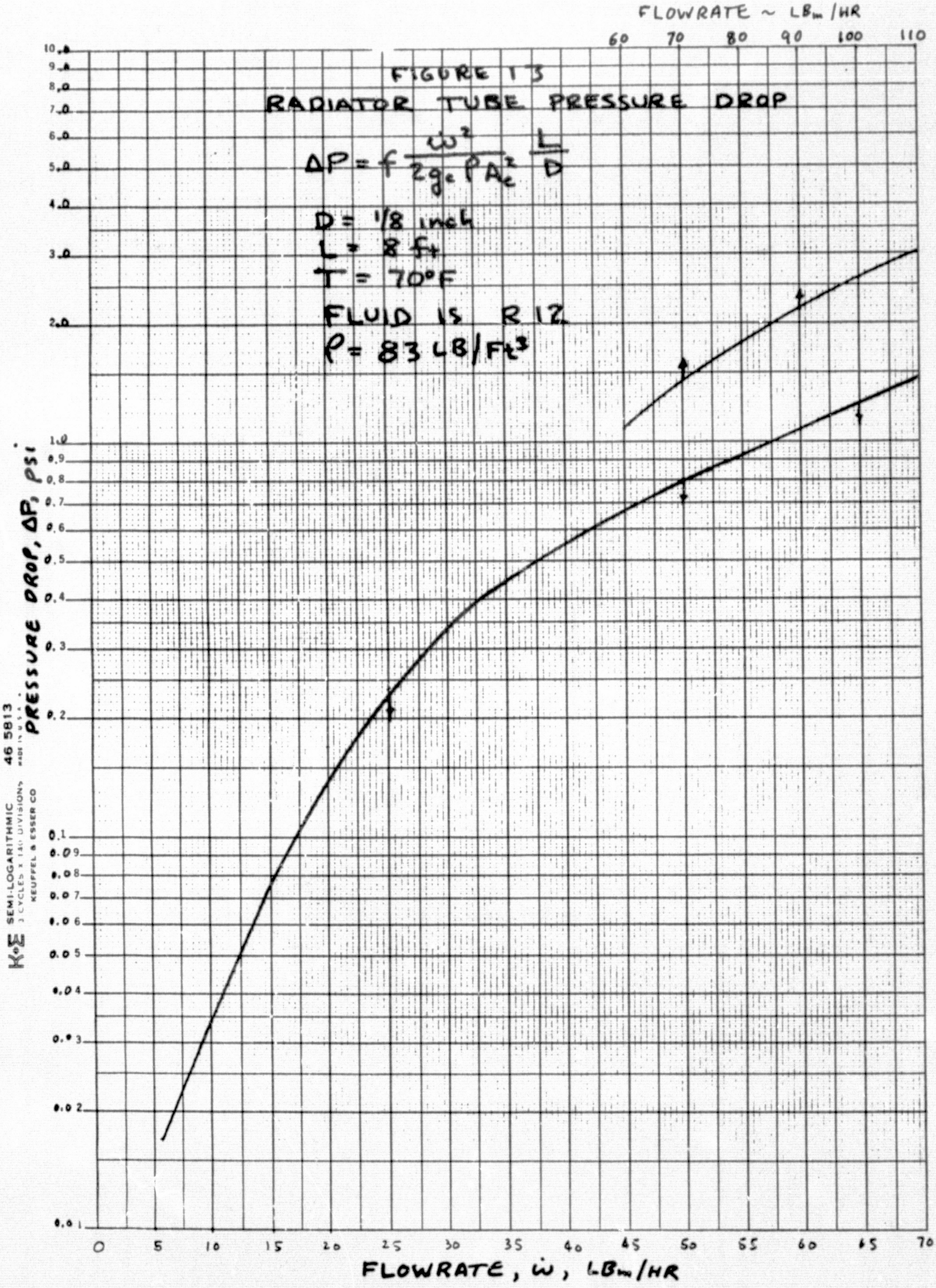


FIGURE 13
RADIATOR TUBE PRESSURE DROP
$\Delta P = f \frac{\dot{w}^2}{2g_c \rho A_c^2} \frac{L}{D}$
D = 1/8 inch
L = 8 ft
T = 70°F
FLUID IS R12
ρ = 83 LB/Ft³
FLOWRATE ~ LBm/HR
PRESSURE DROP, ΔP, PSI
FLOWRATE, ẇ, LBm/HR
K+E SEMI-LOGARITHMIC 46 5813
3 CYCLES X 140 DIVISIONS MADE IN U.S.A.
KEUFFEL & ESSER CO.

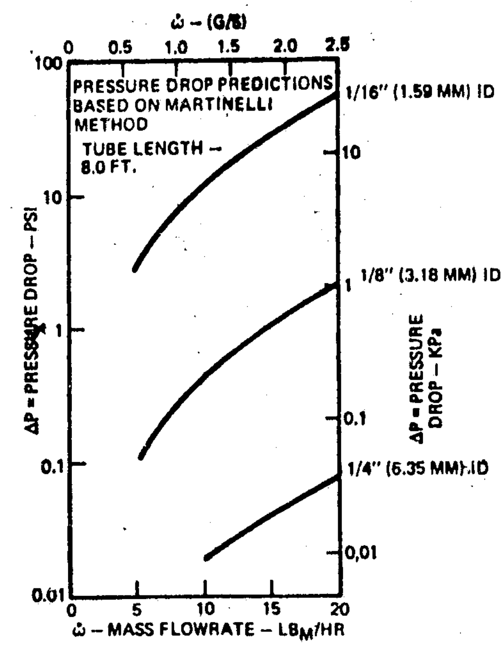


Fig. 14 Radiator/condenser pressure drop versus flow rate

(from Reference [12])

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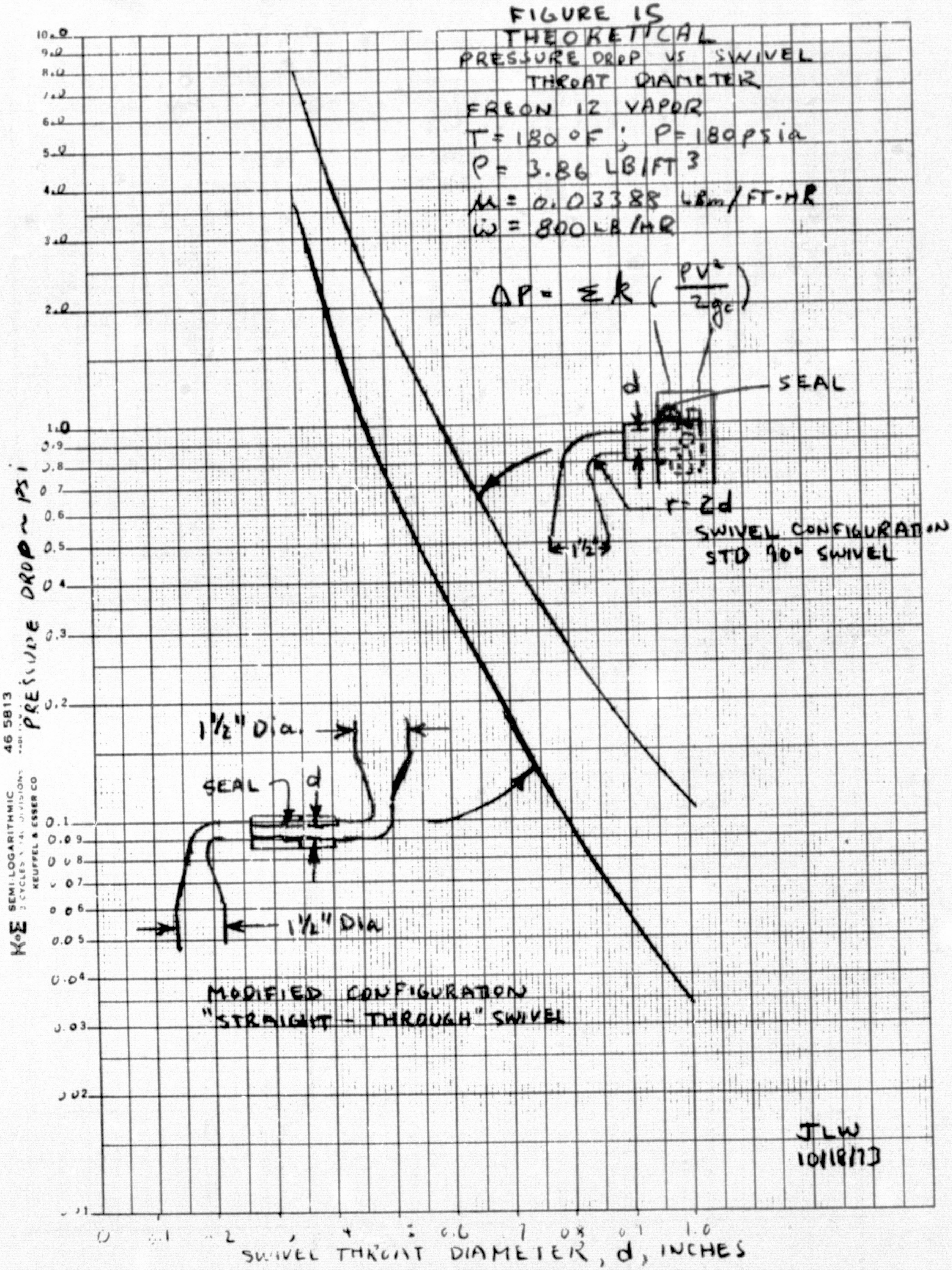


FIGURE 15
THEORETICAL
PRESSURE DROP VS SWIVEL THROAT DIAMETER
FREON 12 VAPOR
T = 180 °F ; P = 180 psia
ρ = 3.86 LB/FT³
μ = 0.03388 LBm/FT-HR
ẇ = 800 LB/HR
ΔP = Σk (ρV²/2gc)
SEAL
d
r = 2d
1½"
SWIVEL CONFIGURATION
STD 90° SWIVEL
1½" Dia.
SEAL
d
1½" Dia
MODIFIED CONFIGURATION
"STRAIGHT - THROUGH" SWIVEL
JLW
10/18/73
PRESSURE DROP ~ PSI
SWIVEL THROAT DIAMETER, d, INCHES
K&E SEMI-LOGARITHMIC 46 5813
KEUFFEL & ESSER CO.

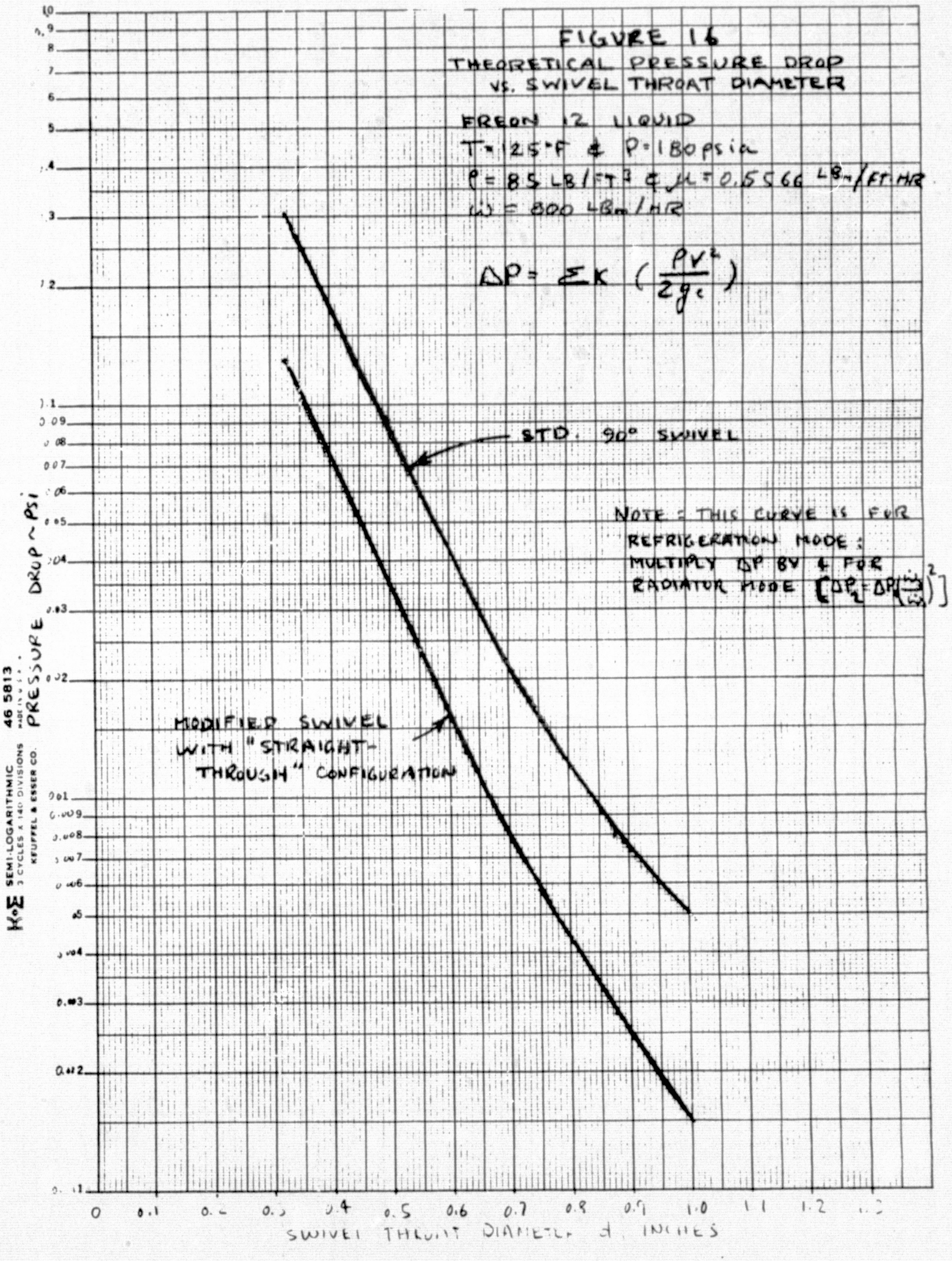


FIGURE 16
THEORETICAL PRESSURE DROP
VS. SWIVEL THROAT DIAMETER
FREON 12 LIQUID
T=125°F & P=180 psia
ρ = 85 LB/FT³ & μ = 0.5566 LBm/FT·HR
w = 800 LBm/HR
ΔP = ΣK (ρV²/2gc)
STD. 90° SWIVEL
NOTE: THIS CURVE IS FOR
REFRIGERATION MODE;
MULTIPLY ΔP BY 4 FOR
RADIATOR MODE [ΔP2 = ΔP1(w2/w1)²]
MODIFIED SWIVEL
WITH "STRAIGHT-
THROUGH" CONFIGURATION
PRESSURE DROP ~ PSI
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0 1.1 1.2
SWIVEL THROAT DIAMETER d, INCHES
K&E SEMI-LOGARITHMIC 46 5813
3 CYCLES X 140 DIVISIONS
KEUFFEL & ESSER CO.

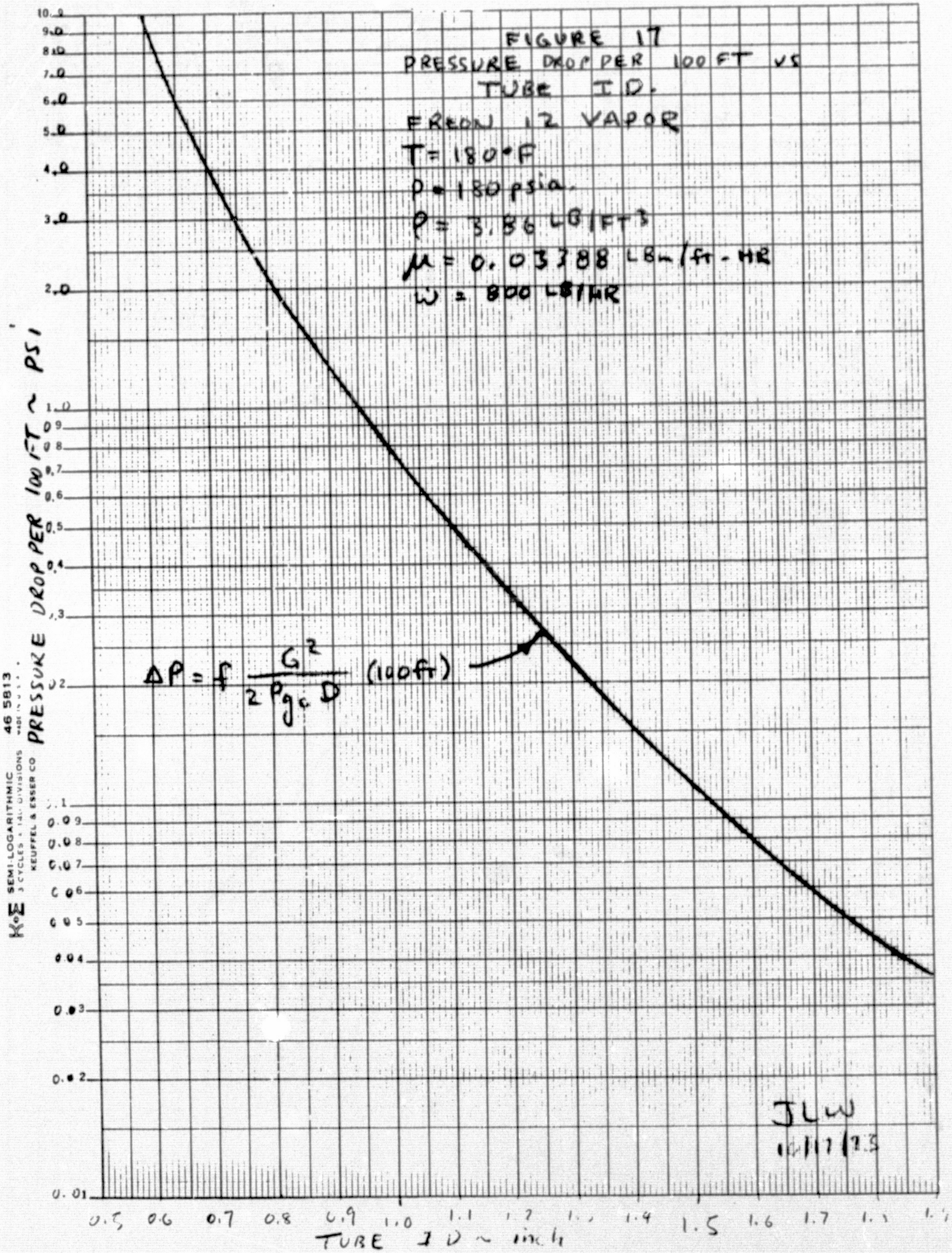


FIGURE 17
PRESSURE DROP PER 100 FT VS TUBE I.D.
FREON 12 VAPOR
T = 180°F
P = 180 psia
ρ = 3.86 LB/FT3
μ = 0.03788 LBm/ft-HR
ẇ = 800 LB/HR
ΔP = f G²/(2ρ g_c D) (100ft)
PRESSURE DROP PER 100 FT ~ PSI
TUBE ID ~ inch
JLW
SEMI-LOGARITHMIC 46 5813
3 CYCLES × 140 DIVISIONS
KEUFFEL & ESSER CO.

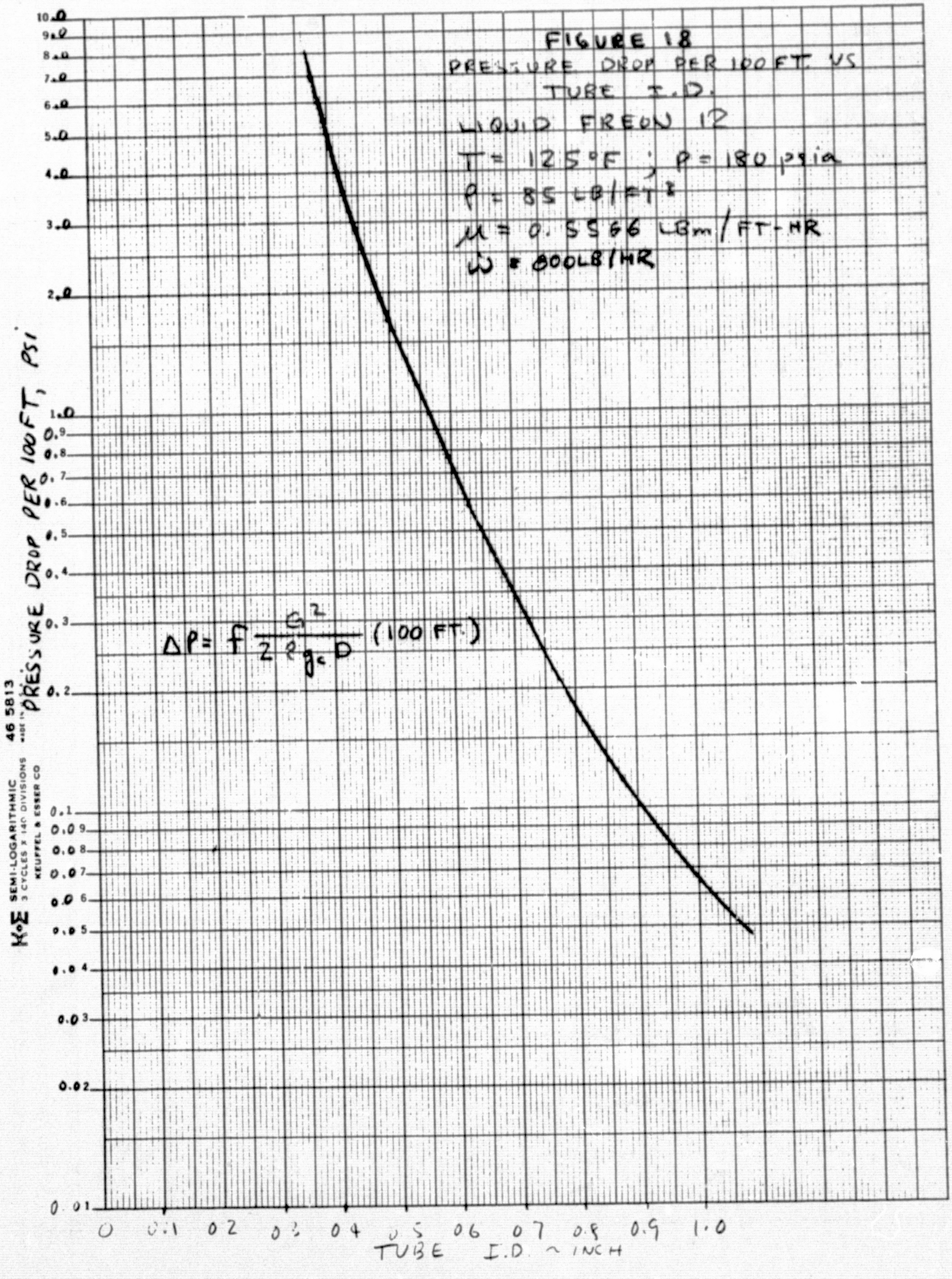


FIGURE 18
PRESSURE DROP PER 100 FT. VS TUBE I.D.
LIQUID FREON 12
T = 125°F ; P = 180 psia
ρ = 85 LB/FT³
μ = 0.5566 LBm/FT-HR
ẇ = 800 LB/HR
$\Delta P = f \frac{G^2}{2 \rho g_c D} (100\ FT.)$
PRESSURE DROP PER 100 FT, PSI
TUBE I.D. ~ INCH
10.0
1.0
0.1
0.01
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
K+E SEMI-LOGARITHMIC 3 CYCLES X 140 DIVISIONS KEUFFEL & ESSER CO. 46 5813

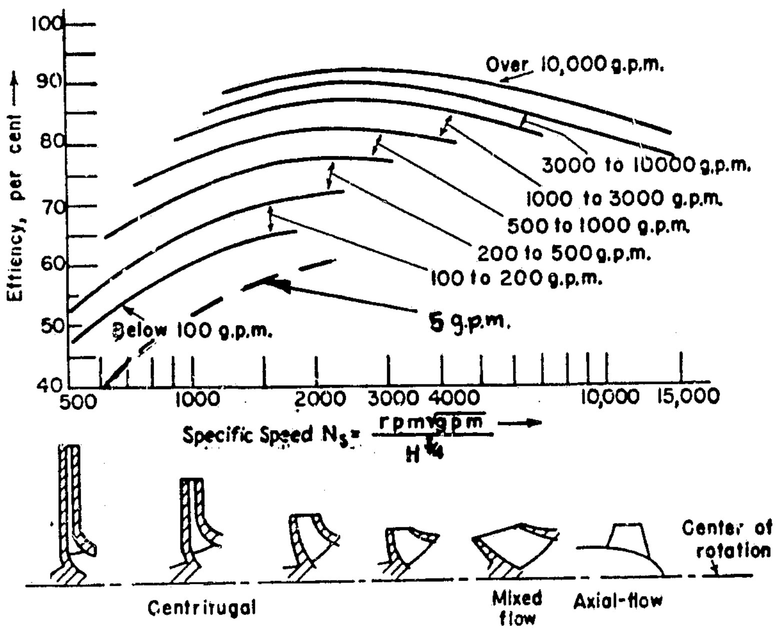


Figure 14 Approximate relative impeller shapes and good average efficiencies obtained for commercial pumps as a function of specific speed.

(From References [17] and [24] - Additional Data from Reference [18])

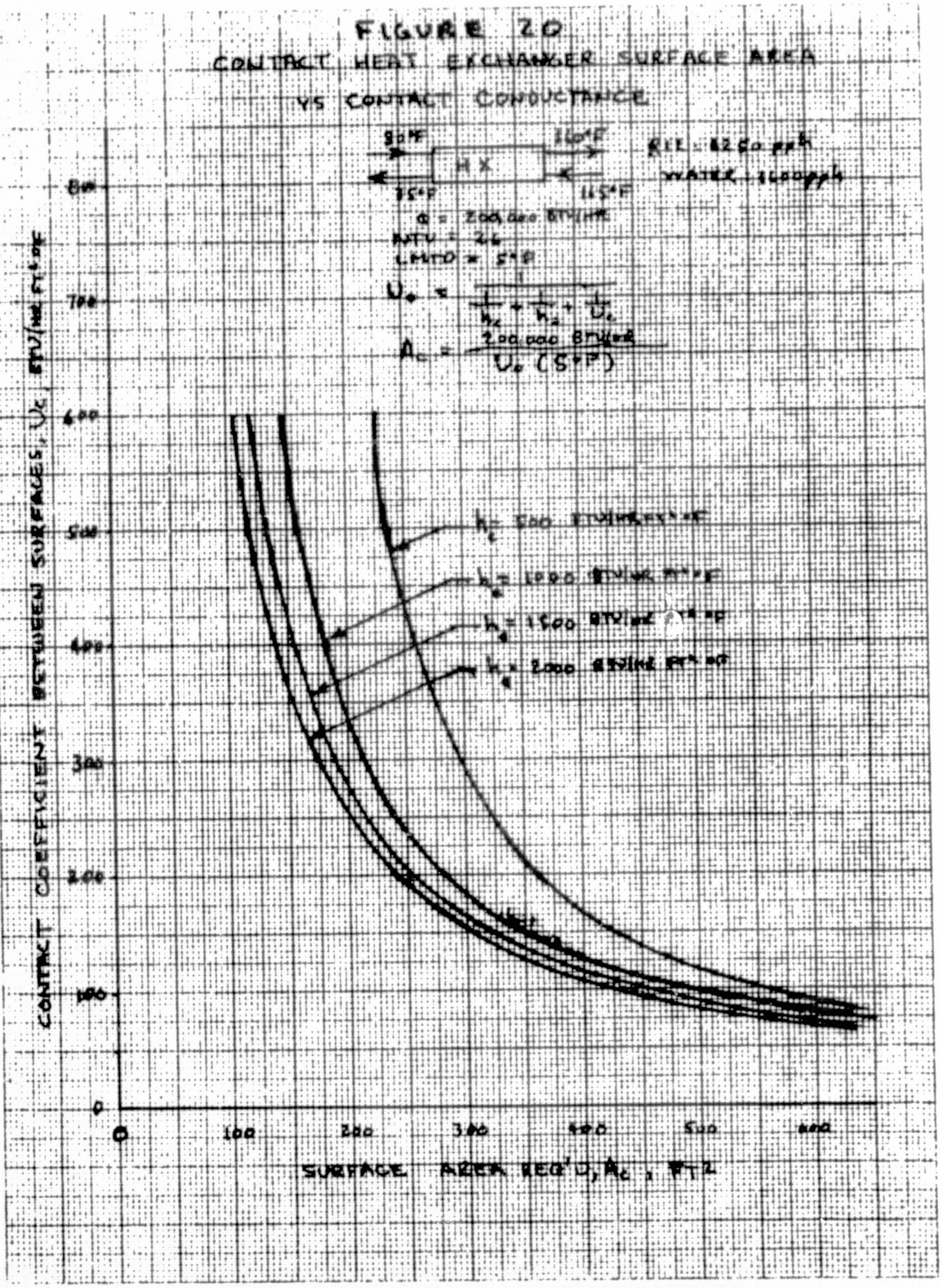


FIGURE 20
CONTACT HEAT EXCHANGER SURFACE AREA
VS CONTACT CONDUCTANCE
80°F
160°F
HX
75°F
115°F
LMTD = 5°F
h_c = 500 BTU/HR FT² °F
h_c = 1000 BTU/HR FT² °F
h_c = 1500 BTU/HR FT² °F
h_c = 2000 BTU/HR FT² °F
CONTACT COEFFICIENT BETWEEN SURFACES, U_c, BTU/HR FT² °F
SURFACE AREA REQ'D, A_c, FT²
0
100
200
300
400
500
600
700
800

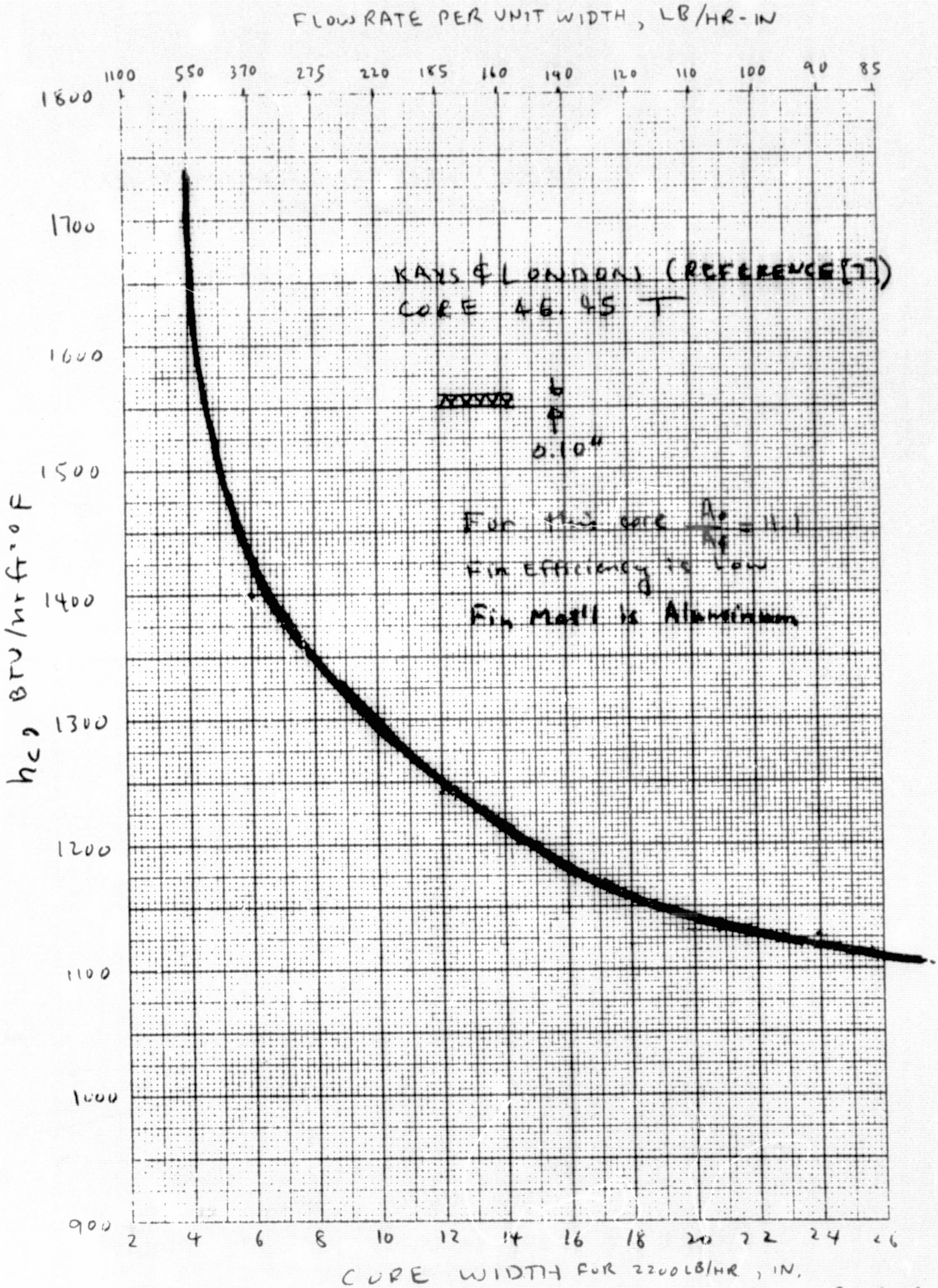


FIGURE 21 HEAT EXCHANGER CORE THERMAL PERFORMANCE

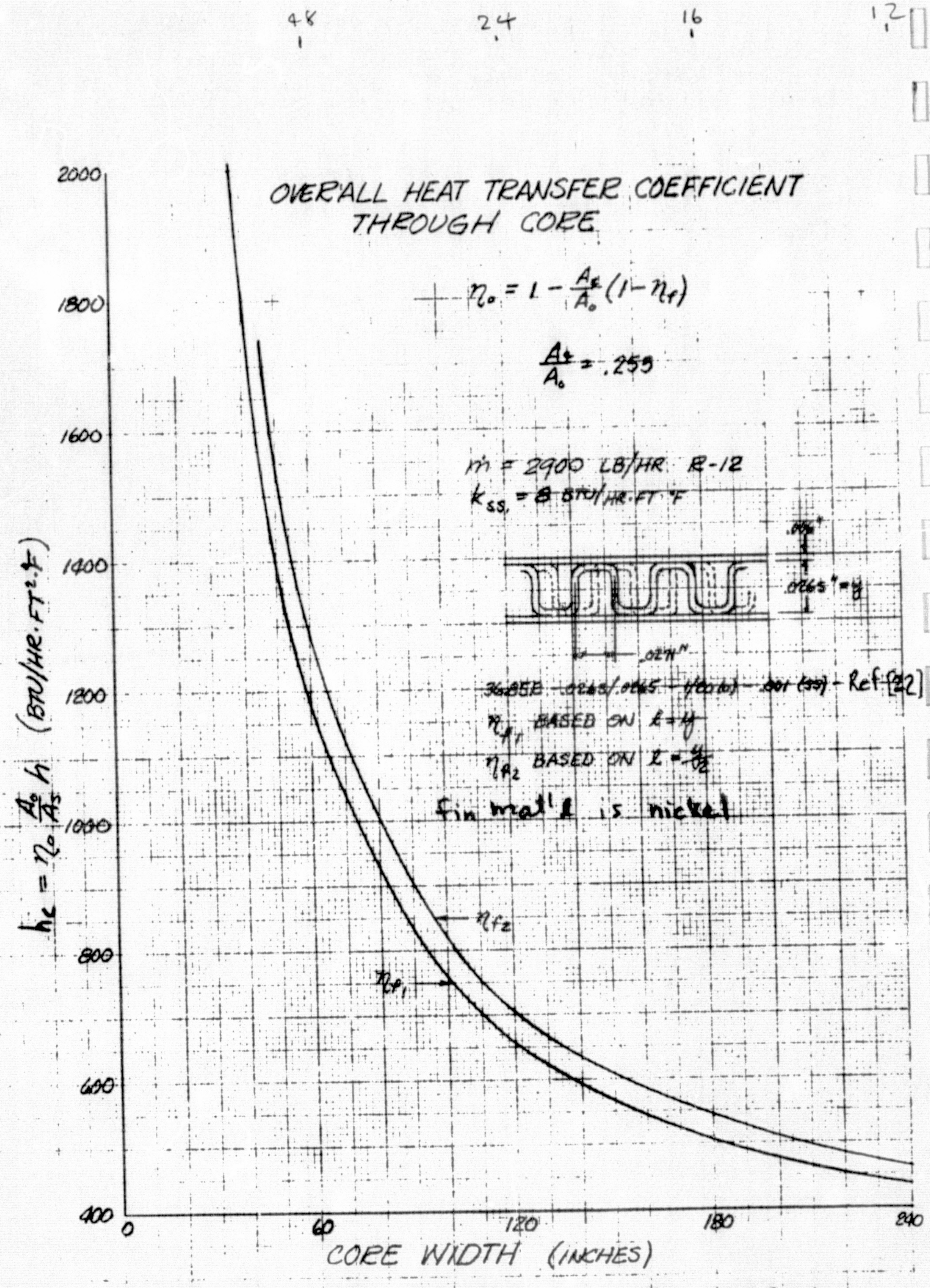


FIGURE 22 VERY COMPACT HX CORE THERMAL PERFORMANCE

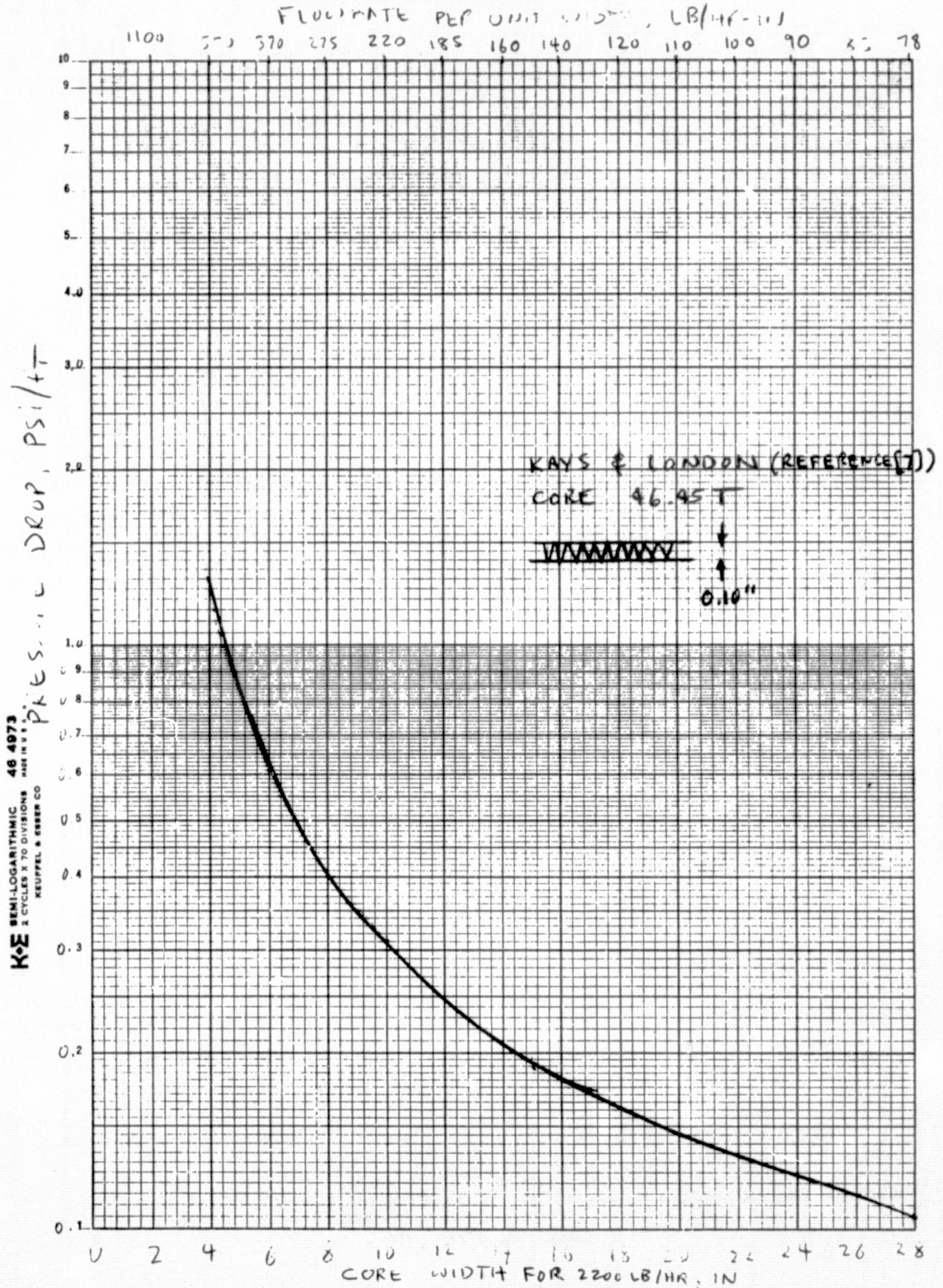


FIGURE 23 COMPACT HX CORE ΔP PERFORMANCE

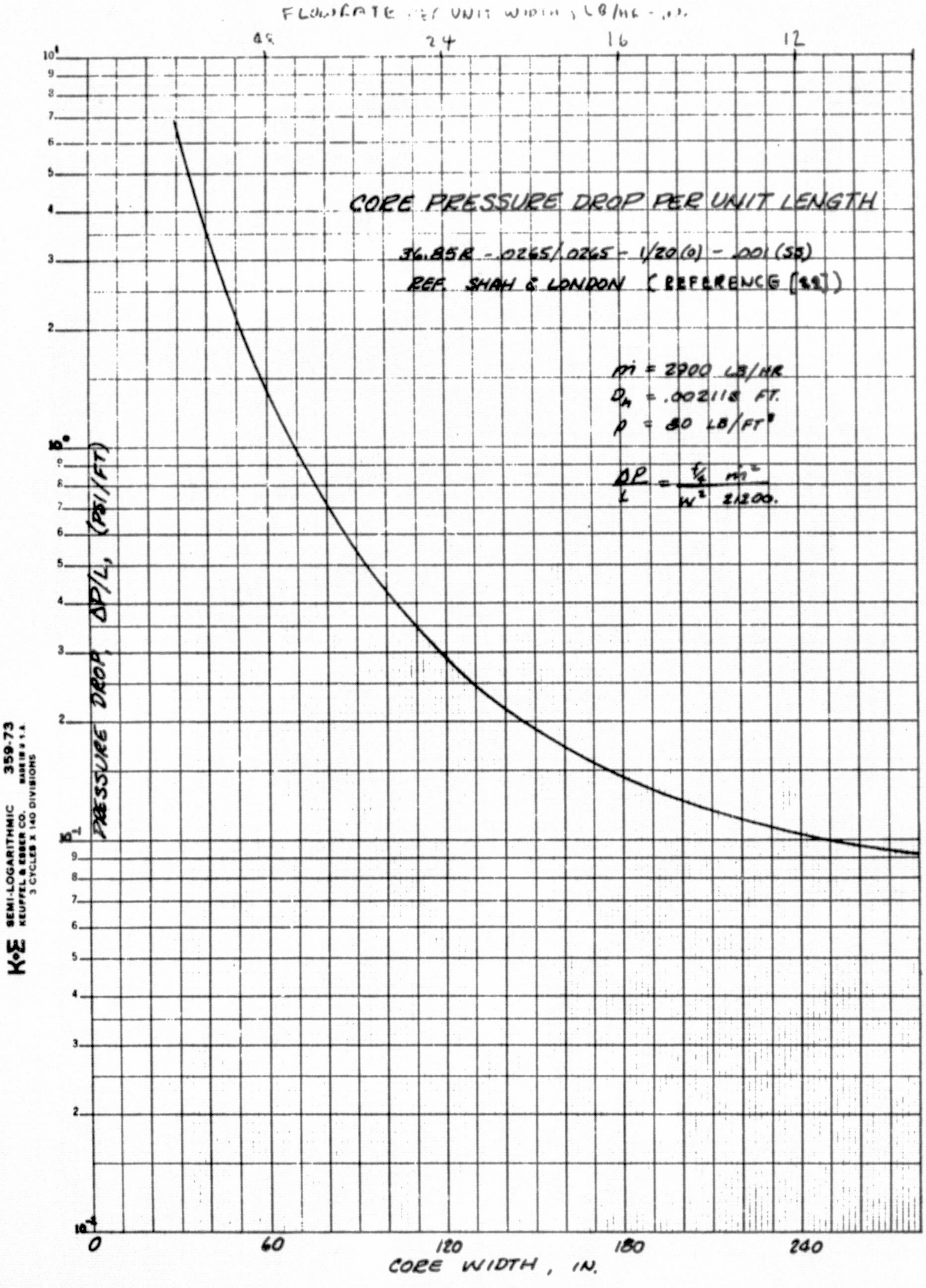


FIGURE 24 VERY COMPACT HX CORE ΔP PERFORMANCE

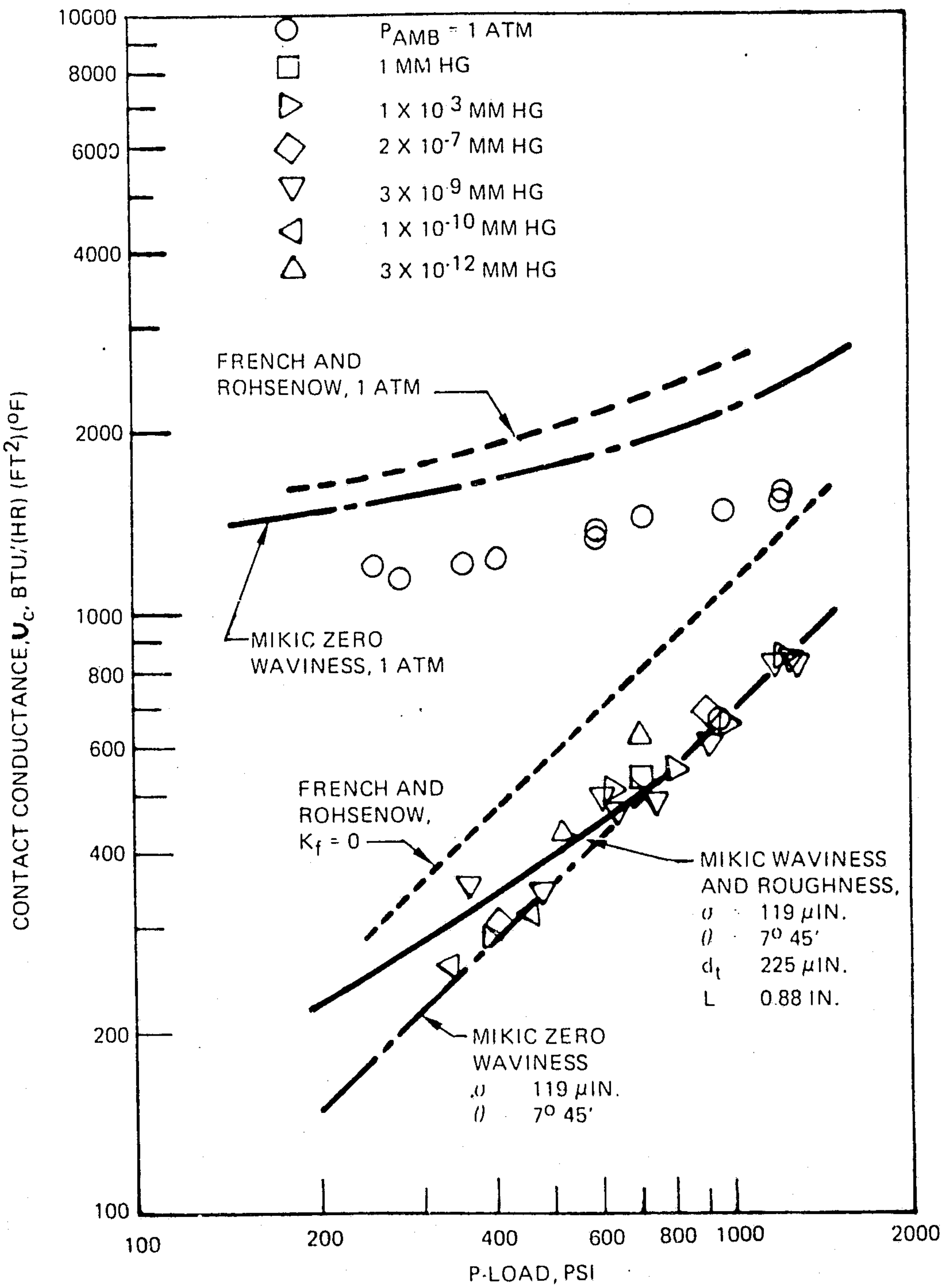
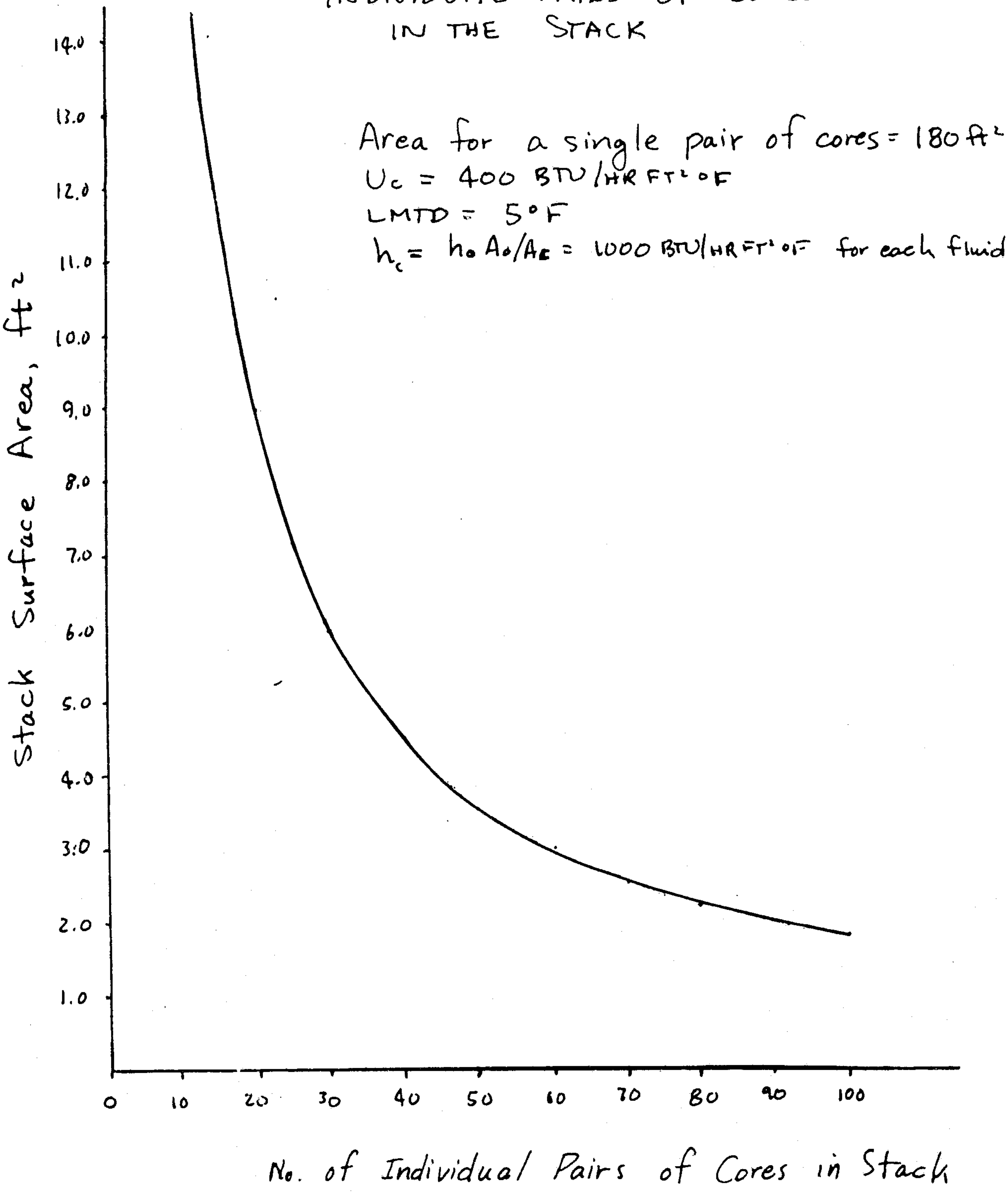


FIGURE 25 THERMAL CONTACT CONDUCTANCE (REFERENCE [21])

FIGURE 26
STACK SURFACE AREA vs. No. OF INDIVIDUAL PAIRS OF CORES IN THE STACK



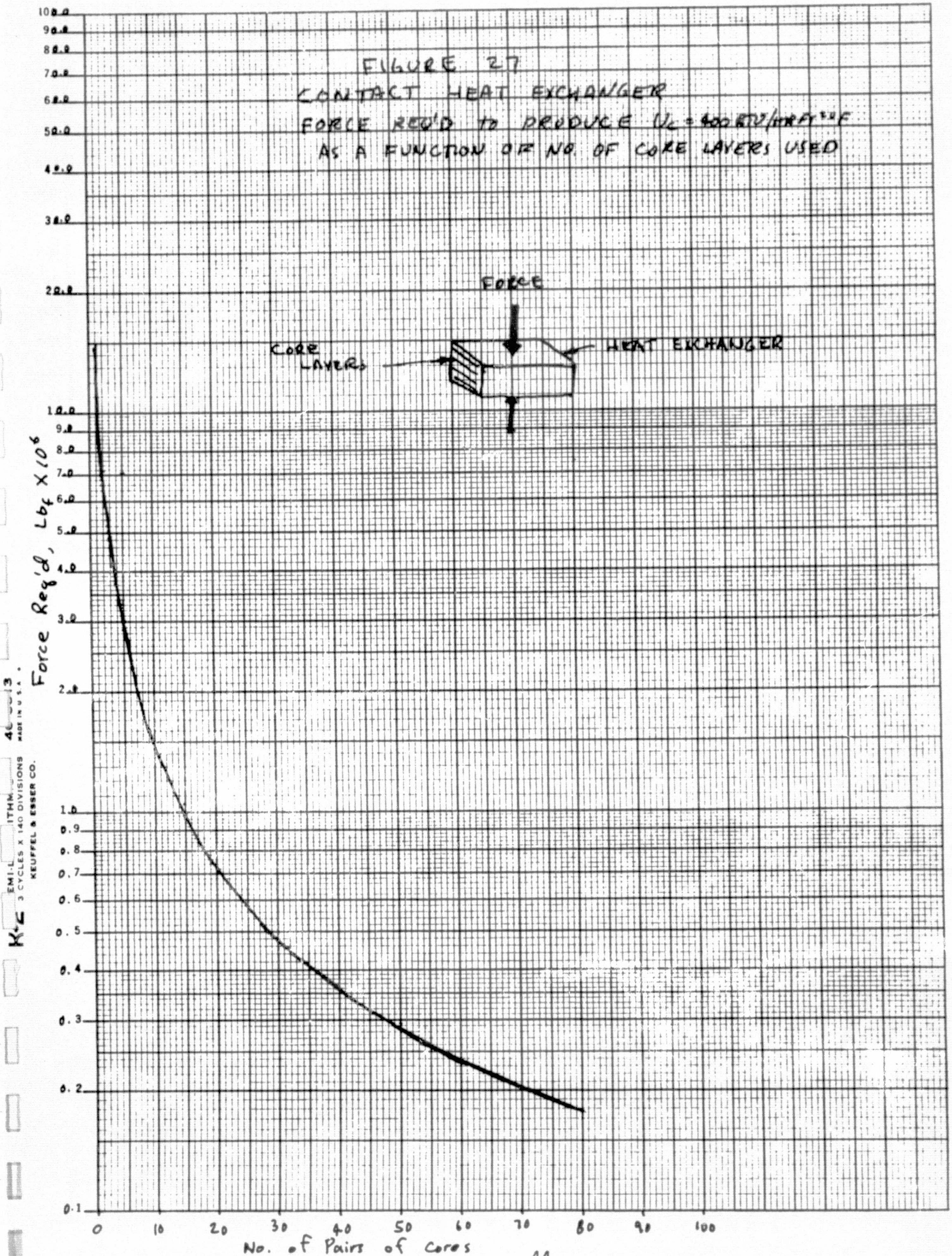
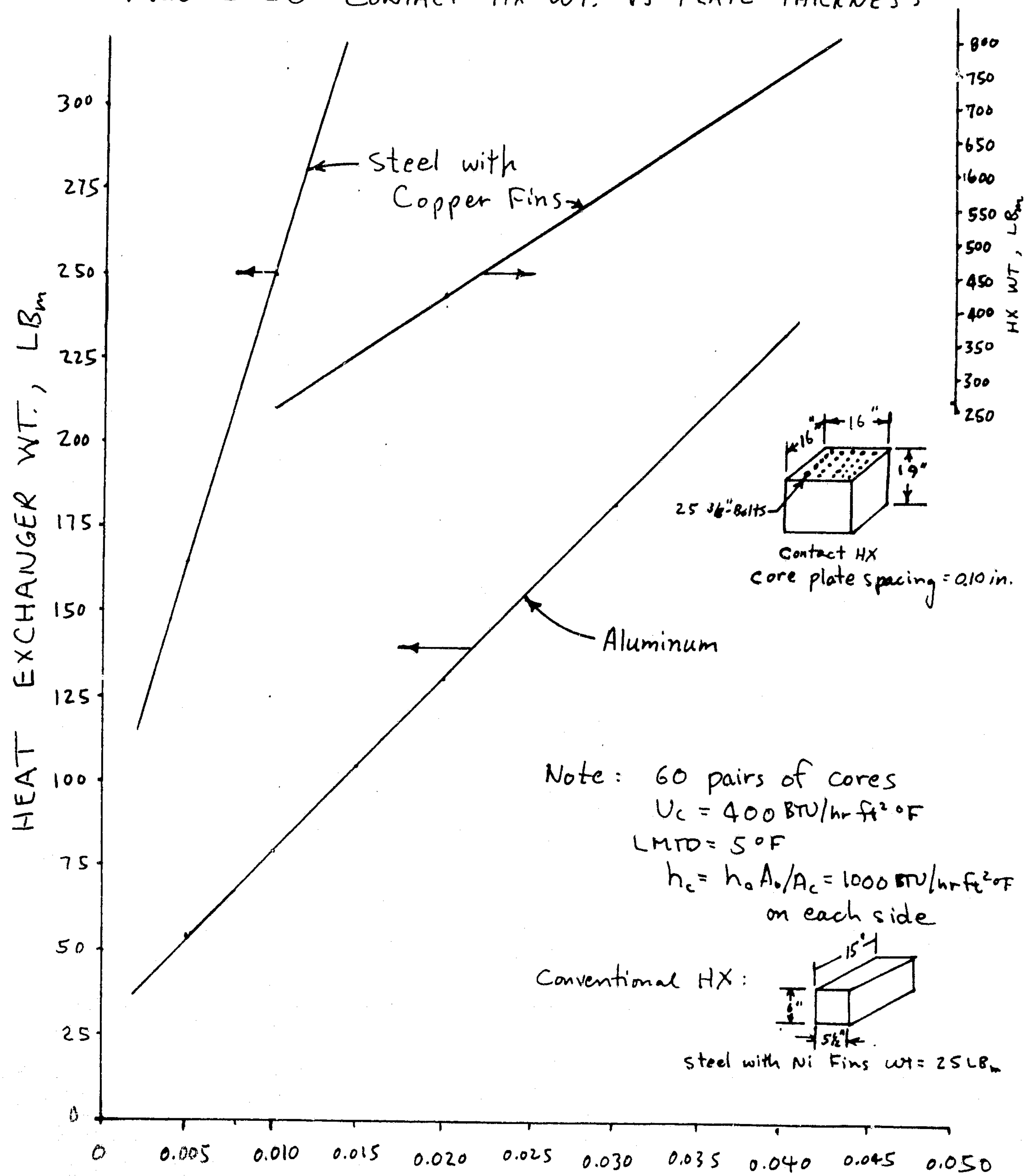
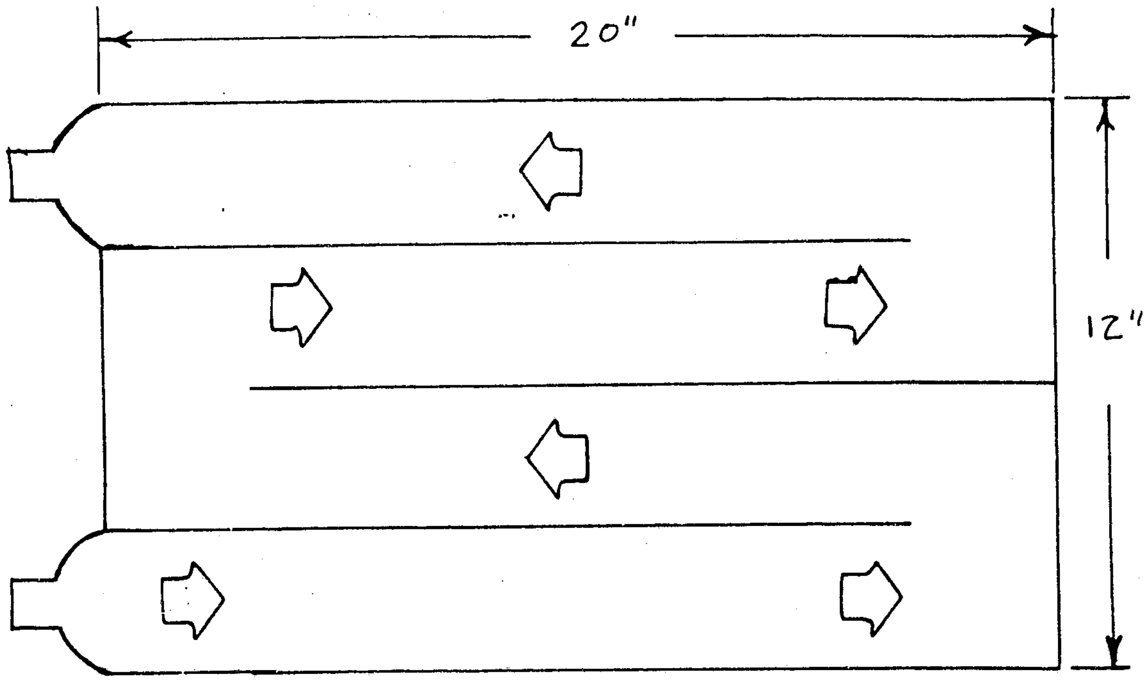


FIGURE 27
CONTACT HEAT EXCHANGER
FORCE REQ'D TO PRODUCE U_c = 400 BTU/HR FT²°F
AS A FUNCTION OF NO. OF CORE LAYERS USED
FORCE
CORE LAYERS
HEAT EXCHANGER
Force Req'd, Lbf X 10⁶
No. of Pairs of Cores
0
10
20
30
40
50
60
70
80
90
100

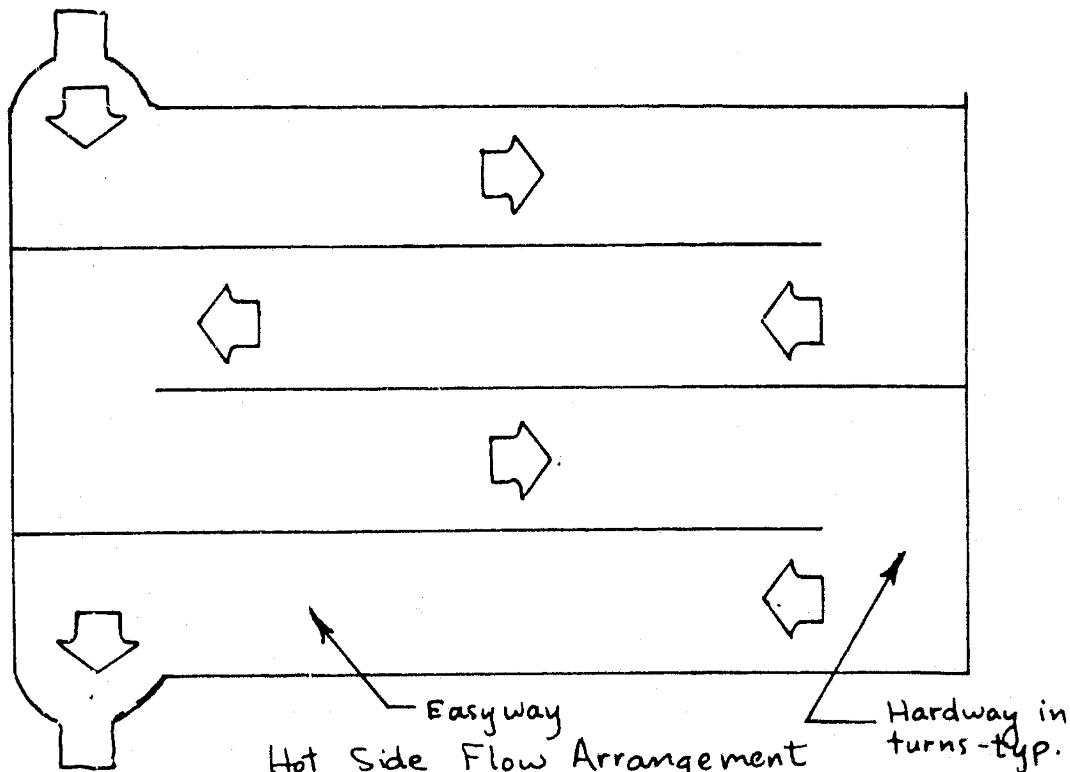
FIGURE 2.8 CONTACT HX WT. VS PLATE THICKNESS





Cold Side Flow Arrangement

- Very Compact HX Core - 36.85R - 0.0265/0.0265 - 1/20(O) - 0.001(SS) Reference [22]. See Figures 22 & 24 for Performance.
- Plate Thickness is 0.015-in.
- 60 core Stack.
- 35 Bolts Press cores Together



Hot Side Flow Arrangement

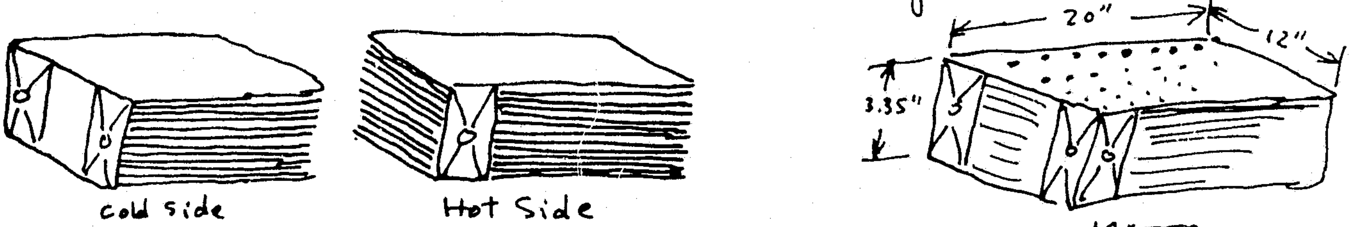


FIGURE 29 Contact HX Design From Very Compact HX Core Mat'l

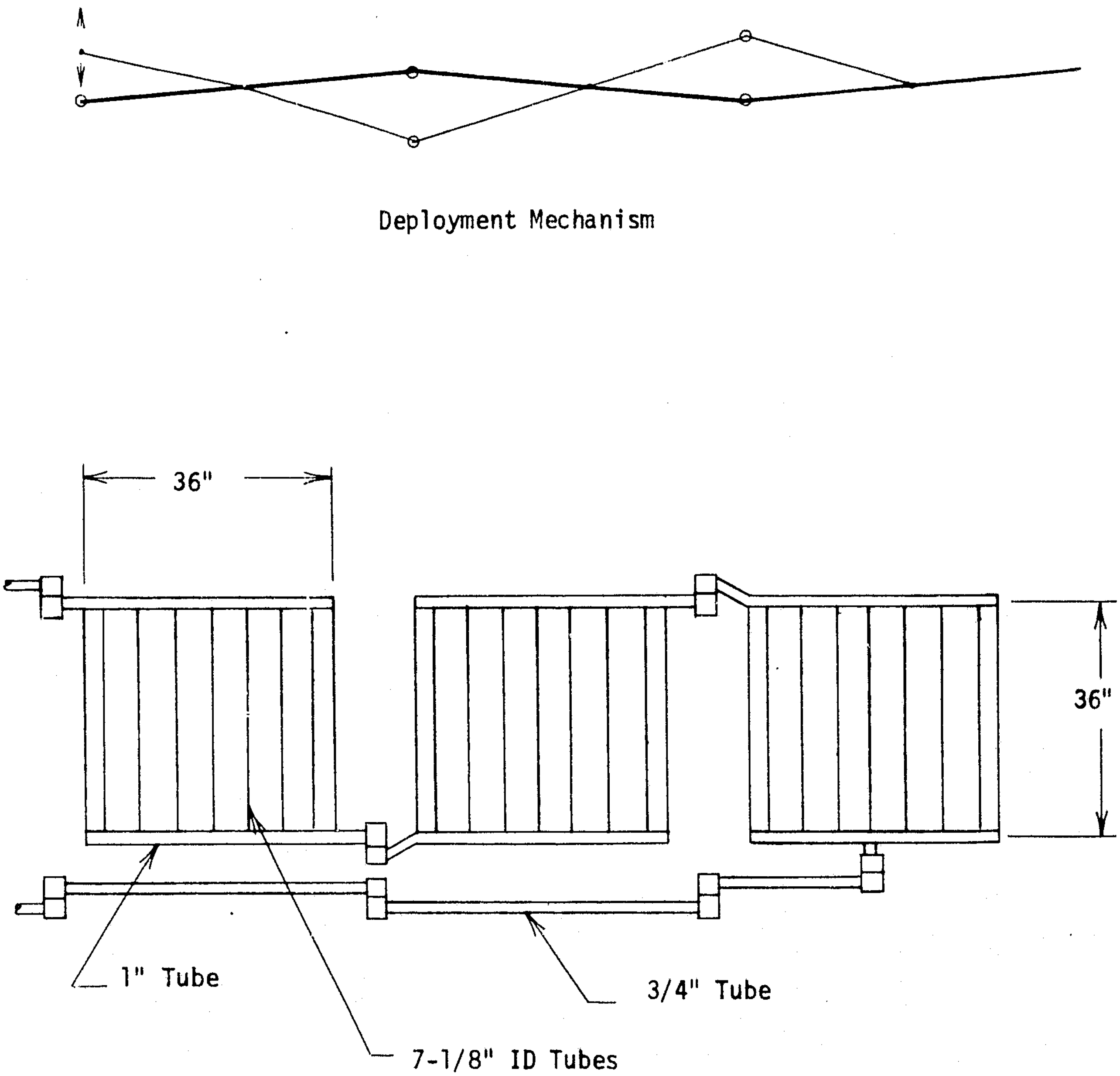


FIGURE 30 LABORATORY PROTOTYPE RADIATOR/CONDENSER DEPLOYMENT MECHANISM AND MANIFOLDS

APPENDIX D

DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE

Contract No. NAS9-13533

Report No. T211-RP-004

PROGRESS REPORT

FOR

PERIOD OF 1-30 NOVEMBER 1973

7 December 1973

Submitted by:

VOUGHT SYSTEMS DIVISION
LTV Aerospace Corporation
Dallas, Texas

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Johnson Space Center
Houston, Texas

Prepared by:

J. L. Williams

J. L. Williams

Approved by:

R. J. French

R. J. French, Supervisor
EC/LS Group

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1.0 INTRODUCTION AND SUMMARY

This fourth monthly progress report under Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 1 through 30 November 1973.

A design review meeting was held at NASA-JSC on 19 November at which time preliminary designs of flight and laboratory prototype Self-Contained Heat Rejection Modules (SHRM) were presented. After consideration of alternatives, NASA-JSC elected to have VSD fabricate a breadboard SHRM using flight-type flow equipment. This is in lieu of building a laboratory prototype with a deployable radiator system, which would not use flight-type flow equipment. This decision was principally based on the selection of the ATM solar cell array deployment mechanism as the concept for deploying SHRM radiators. Since this deployment system was used on Skylab, it was deemed reasonable to shift the emphasis on SHRM from the radiator deployment mechanism to flight prototype equipment.

Work planned for next month includes design of the breadboard SHRM, preparation of procurement specifications, and placing of orders for hardware.

2.0 DISCUSSION OF WORK PERFORMED THIS MONTH

2.1 Preliminary Design Review

The preliminary design review for SHRM was held at NASA-JSC on 19 November 1973. The Enclosure to this report contains the presentation made at that time. It contains a review of concept evaluations, and parametric analyses used in establishing the design. The flight prototype design incorporates a deployment mechanism similar to that used on the Apollo Telescope Mount (ATM) Solar Cell Array. Radiator panel and fluid flow hardware designs which are compatible with the ATM deployment mechanism were rendered. In addition flow equipment and hardware were sized for the SHRM design heat loads. The flight prototype design incorporates four wings with four radiator panels each for space station applications (including stations deployed from shuttle), and two wings with four radiator panels each for space shuttle payload application.

A design was also rendered for a small-sized (not thermal-scale modeled) version of the flight prototype system. This laboratory prototype SHRM included a scissor-type deployment mechanism which would deploy three radiator panels in the VSD Space Environment Simulator. The small heat rejection capability of the resulting system results in a mismatch between the required flowrates and the rotating machinery to be used in the flight prototype SHRM. Use of full-scale rotating machinery would involve complicated partitioning of flow between test support equipment and the prototype system or operation at flowrates far from the design points, or possibly both. This coupled with the fact that the selected radiator deployment mechanism had been previously used in space (on Skylab) led to the decision to build a breadboard system from flight-type hardware. This equipment could then be used in flight prototype testing which is under consideration by NASA. The breadboard tests at VSD will use a single radiator panel which was originally fabricated by VSD for another program. This panel is very similar to the SHRM design, and may be modified by reducing the length to more nearly conform with the SHRM design. This radiator panel will incorporate swivels which can be flexed on the inlet and outlet lines. The breadboard system will also incorporate an accumulator with flight prototype controls, and a feasibility demonstration type contact heat exchanger (not a flight prototype).

2.2 Breadboard System Equipment Sizing

Sizing for the breadboard SHRM has been initiated. The equipment will be sized to handle the flow required for one wing (i.e., four radiator panels) of the total system. This is the full size that would be carried on the shuttle for payload heat rejection, and is capable of rejecting up to 40,000 BTU/hr (14.65 kw). Only the dual-mode wing will be considered : of course it includes both vapor cycle and conventional radiator operation. The high-temperature radiator wing is the same as the dual-mode wing in the conventional radiator mode, except that it may use two-D radiator panels rather than straight parallel flow tubes.

2.2.1 Radiator Panel

The existing radiator panel has the following characteristics:

material	:	aluminum
width	:	72 in. (1.83 m)
length	:	144 in. (3.66 m)
thickness	:	0.063 in. (1.6 mm)
tube I.D.	:	1/8-in. (3.175 mm)
No. of tubes	:	9
tube spacing	:	8-in. (20.32 cm)
surface coating	:	black velvet paint ($\varepsilon \approx 0.9$)

Plans are to reduce the length of the panel to 96-in. (2.44 m). This panel was designed for a fin efficiency of 0.9,in an application requiring one side operation. For two-sided operation in the condensing mode, the panel will have a fin efficiency of about 0.84. Heat rejection in the SHRM design environment would be 10,110 BTU/hr (2.96 kw).

2.2.2 Pump

The flowrate required to the breadboard SHRM radiator panel is 610 lb/hr (77 g/s), and the flowrate to an entire wing of the SHRM flight prototype is 3125 lb/hr (393 g/s.).

The pressure drop requirement for the clean system is around 20 psi (137.8 kN/m^2), however, for use in testing additional pressure drop will be induced by instrumentation, so a pump pressure drop of 30 psi (206.7 kN/m^2) was selected.

This results in a pump specific speed of 1274 (based on a pump speed of 11,800 rpm) and a centrifugal pump efficiency of 55%. Power required for the pump will be around 170 w.

2.2.3 Compressor

For the vapor cycle mode the evaporator heat load is

$$q_L = \frac{q_r}{(1 + \frac{1}{COP})} = \frac{50{,}000 \text{ BTU/hr}}{(1 + \frac{1}{2.5})} = 35{,}714 \text{ BTU/hr (10.46 kw)}$$

This is approximately a 3-ton unit. Fairchild-Stratos makes a heli-rotor compressor in the 3-ton capacity range; however, the availability of the unit has not been verified.

2.2.4 Accumulator

The accumulator should be 650 cu.in. (10,650 cc); however preliminary indications are that it may be difficult to obtain one in this range. A much smaller accumulator could be used in the breadboard system. The 300 cu.in. (4900 cc) unit identified in the Enclosure would be satisfactory.

2.3 Refrigeration Operation

SHRM has a requirement to be capable of providing a 0°F (255.5°K) heat sink in addition to the normal 40°F (277.8°K) sink. Optimization of this system is difficult with no power penalty. Figure 1 shows the heat load which can be rejected from a single wing of SHRM (four panels, both sides radiating) and power required as a function of condensing temperature. Based on this figure, it would seem probable that a condenser temperature of 60°F (288.9°K) would be selected. This gives a refrigerated load of 5.2 kw and a power requirement of 1.3 kw.

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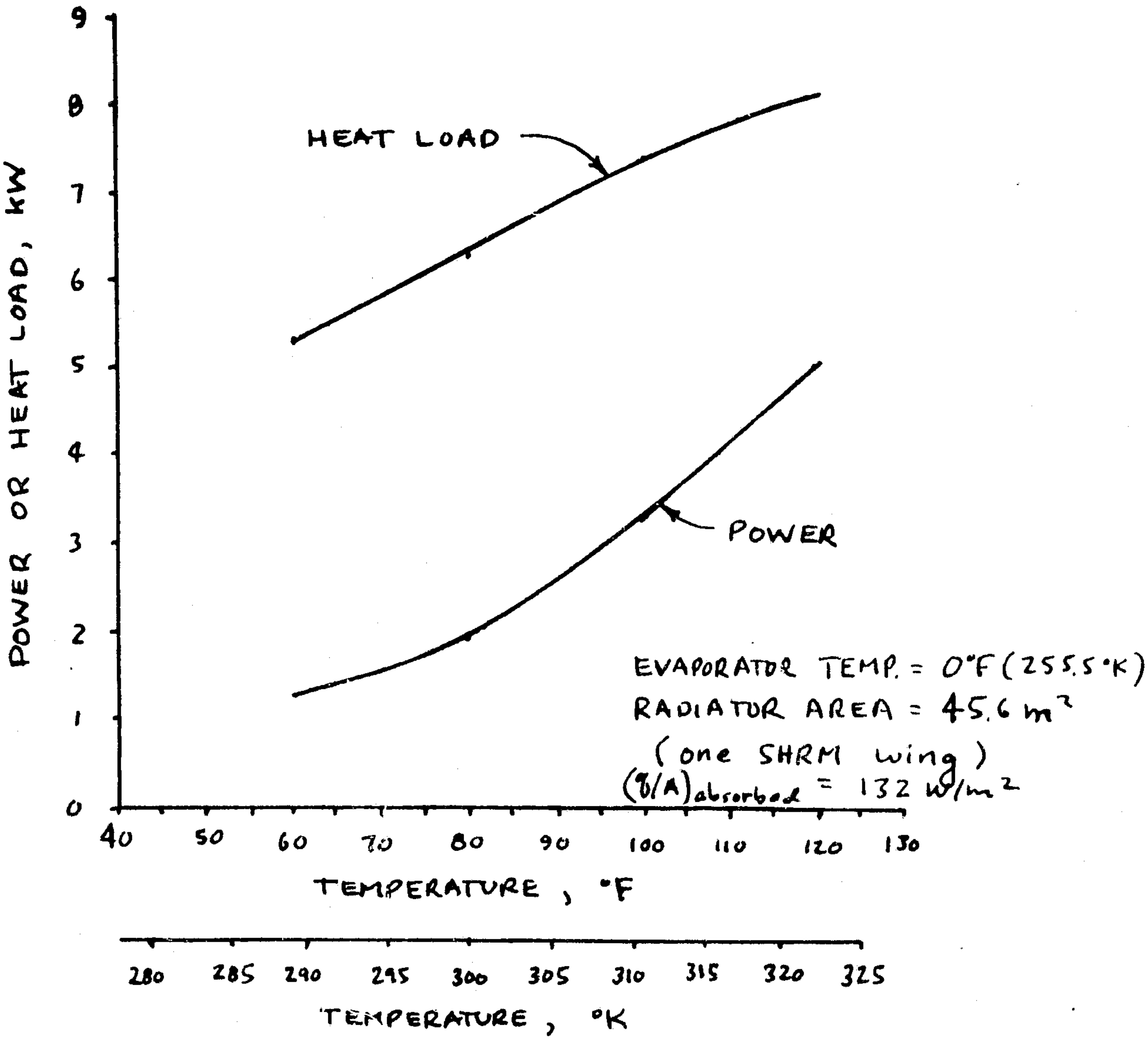

FIGURE 1 SHRM REFRIGERATOR PERFORMANCE

3.0 CURRENT PROBLEM AREAS

There are no current technical problem areas. The program is on schedule as indicated in the figure on page 4 of the enclosure to this report. The program is entering into a crucial phase with regard to schedule; this is because the lead times for the selected system components may not match those anticipated in the schedule.

4.0 WORK PLANNED FOR NEXT MONTH

The work planned for next month includes completion of sizing of the SHRM breadboard components, and initiation of procurement activity.

APPENDIX E

DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE

Contract NAS9-13533
Report No. T211-RP-005

PROGRESS REPORT FOR PERIOD
1 December 1973 through 21 January 1974

22 January 1974

Submitted by

VOUGHT SYSTEMS DIVISION
LTV Aerospace Corporation
Dallas, Texas

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Johnson Space Center
Houston, Texas

Prepared by:

J. L. Williams
J. L. Williams

Approved by:

R. J. French
R. J. French, Supervisor
EC/LS Group

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1.0 INTRODUCTION AND SUMMARY

This fifth monthly progress report under Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 1 December through 21 January 1974.

The effort during the past month was spent in sizing equipment to be purchased for the breadboard test under the guidelines established in Reference [1]. Informal quotes have been solicited for all major components, and have been received for all except the contact heat exchanger and the pump. These inquiries indicate that the total cost of materials for system components will exceed the budget for material, including credit for the budget originally allocated for fabrication of laboratory prototype radiator panels and deployment mechanism. It is recommended that purchase of some components be deferred until the flight prototype system test phase of the program.

2.0 DISCUSSION OF WORK PERFORMED THIS MONTH

2.1 <u>Analysis</u>

The analysis effort has included sizing of equipment for use in the breadboard test. This equipment is sized to be suitable for use in prototype systems testing to the greatest extent possible.

2.1.1 Breadboard System Schematic

The heat rejection capacity for the breadboard system is limited by the VSD space environment simulator (SES) to about 6 kw (20,000 B/H). The desire to use flight prototype equipment which is designed for a heat rejection of 14.65 kw thus results in a complication of the breadboard test. Figure 1 shows the schematic for the breadboard system test. As indicated, auxiliary heat rejection is required to permit system operation at the higher capacity. Control of the auxiliary cooling system is the most significant complicating factor in the test. Control is achieved by use of a manual flow control valve, which is set based on real-time data read-outs.

2.1.2 Contact Heat Exchanger

The contact heat exchanger has been optimized using the VSD Heat Exchanger Design Routine (HEDR). Sizing was based on the intercooler mode of operation, which occurs when the system is in the conventional radiator mode of operation. This mode was used for design because the log mean temperature difference is considerably less in this mode than in the condensing mode (5.55°k vs 18°K), and the cold side (i.e., the radiator side) heat transfer coefficient is smaller in single phase operation than it is in condensing flow. The design conditions were:

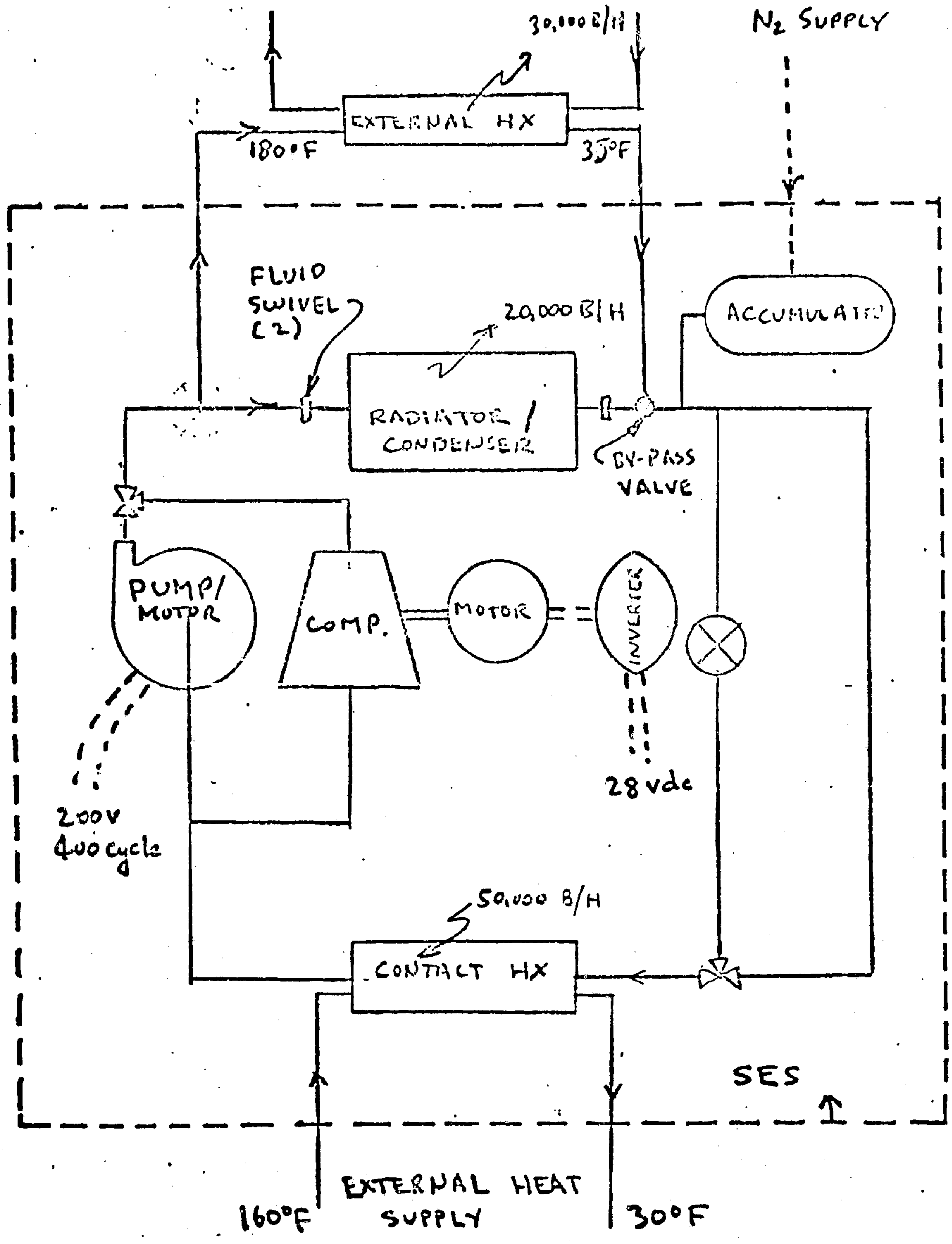

FIGURE I SHRM BREADBOARD TEST SCHEMATIC

JLW
12/14/

$Q = 14.65$ kw (50,000 B/H)

Cold Side Fluid; R12

Hot Side Fluid; Water

Cold Side Temps; $T_{in} = 272.2°K$; $T_{out} = 308.3°K$

Hot Side Temps; $T_{in} = 313.9°K$; $T_{out} = 277.8°K$

Cold Side Flowrate; 0.201 kg/s (1600 lb/hr)

Hot Side Flowrate; 0.097 kg/s (770 lb/hr)

Cold Side Pressure Drop; 8.62 kN/m^2 (2.5 psi)

Hot Side Pressure Drop; 3.45 kN/m^2 (0.5 psi)

The core characteristics on which the design was based are:

Type; Lanced Fin

Plate Spacing; 2.54 mm (0.1-in.)

Fin Thickness; 0.127 mm (0.005-in.)

Fin Pitch; 7.28 fin/cm (18.5 fin/in)

Hydraulic Diameter, 4 r_h; 1.613-mm (0.005292 ft)

Total Heat Transfer Area/Vol. Between Plates, β; 2130 $\frac{m^2}{m^3}$ (650 $\frac{ft^2}{ft^3}$)

Plate Thickness; 0.508-mm (0.020-in.)

The design is based on a contact conductance of 2271 J/m^{2}S°K (400 B/H-ft^2-°F

The resulting design has the characteristics:

Length = 97.8 cm (38.5-in.)

Width = 38.35 cm (15.1-in.)

Volume = 1309 cm^3 (0.0462 ft^3)

The design has 10 plates (with a single flow passage) on both the hot and cold sides in the arrangement shown on Figure 2. This particular design is to be bolted together to obtain the necessary contact force. The same coldplate configuration is used on each side to reduce costs. The size of the coldplates and the resulting heat exchanger are reasonable. The manufacture of the individual cold plates requires machining of both surfaces of the cold plates to produce a surface finish with a crest to-trough roughness of 0.001 to 0.004 mm, based on literature data on contact conductance (References [2] and [3]). The cost of coldplates such as these is roughly estimated at \$500 to \$1000 each, for a total cost of \$10,000 to \$20,000 for the contact HX.

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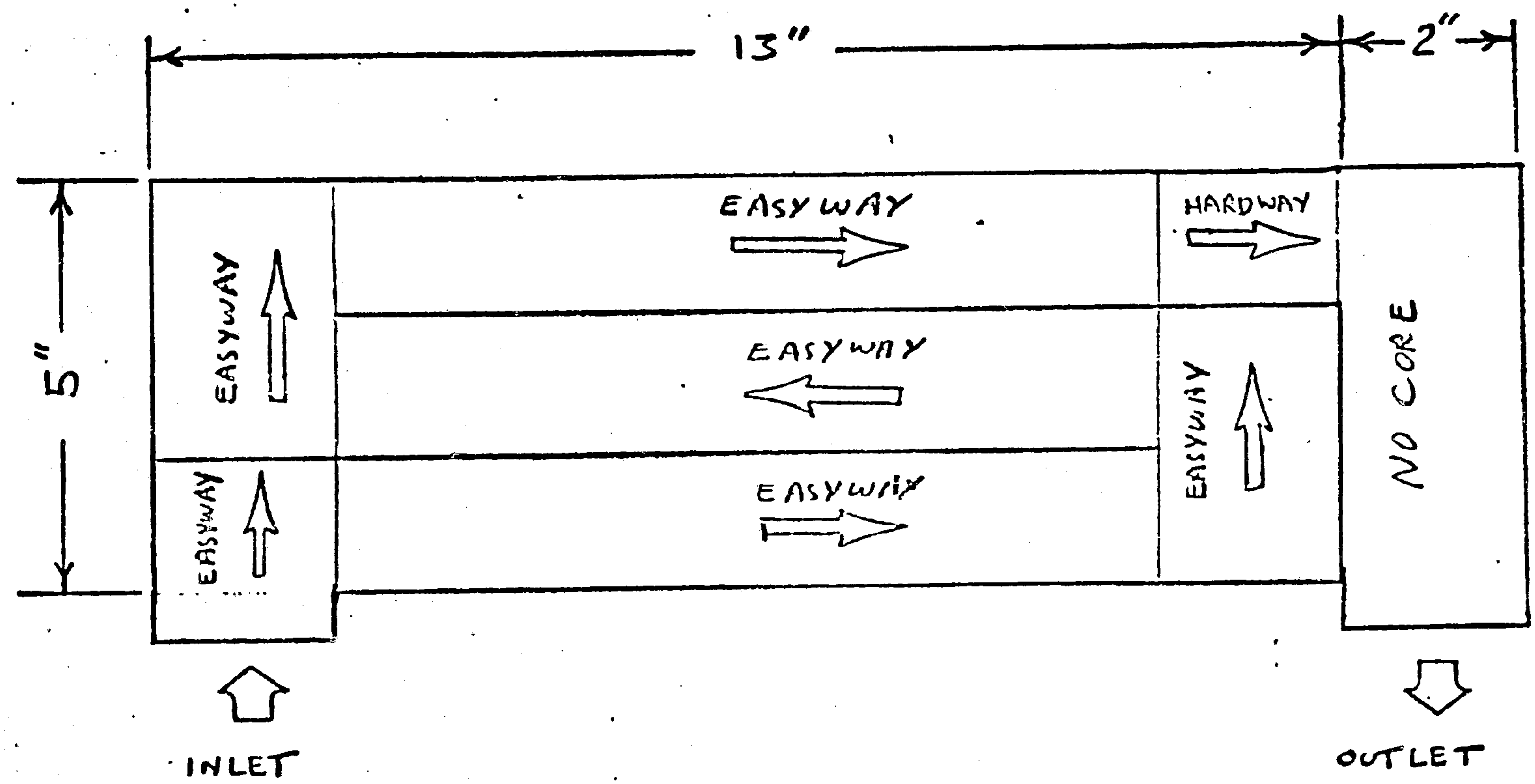

TYPICAL FOR 20 COLD PLATES

ASSEMBLY : 10 COLDPLATES ARE USED ON EACH SIDE OF HX

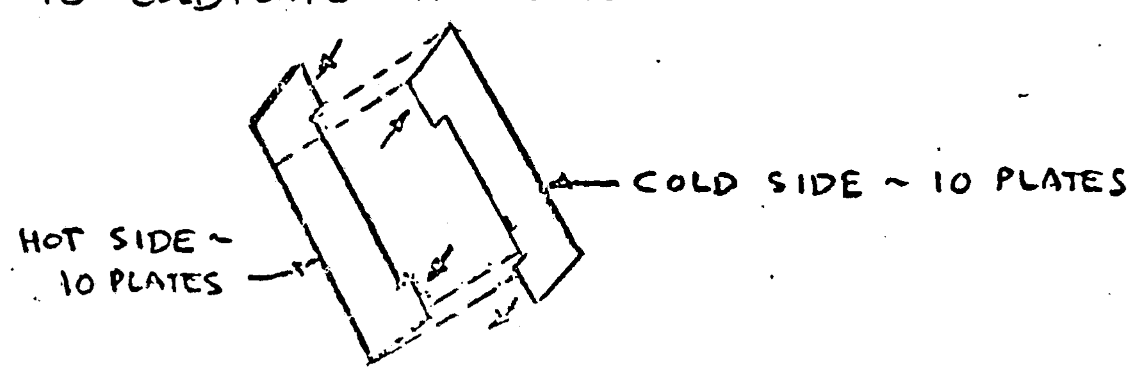

FIGURE 2 : 50,000 B/H CONTACT HX DESIGN

Areas of uncertainty in the design of the contact HX include; (1) the surface roughness required, (2) the optimum thickness of the plates, (3) the force required on the plates, and (4) the distribution of the force on the coldplate. There is enough uncertainty in these factors to make the investment of $10,000 or more in the contact HX questionable without any development testing experience to back up the design. For that reason it is proposed that a development program be undertaken at this time to explore the areas of uncertainty discussed above. This will necessitate the use of a conventional HX in the breadboard SHRM test.

2.2 Component Procurement Activity

The SHRM component procurement activity is summarized in Table I, which includes component description, vendors contacted, approximate cost, lead time, and comments. The accumulator is a troublesome item, since it is very important to the test, and it has a very long lead time and is expensive. VSD is exploring alternatives, such as fabricating the unit in-house, or using a bladder-type accumulator rather than the bellows-type.

Generally these approaches do not appear to be attractive, however, VSD may be able to reduce the lead time by fabricating in-house if a suitable bellows can be found. The only advantage would probably be in lead time reduction, since it is unlikely that sufficient money can be saved to off-set the risk in this approach. A bladder-type accumulator is satisfactory if a suitable bladder can be found; however, the ones which have been identified so far have not been compatible with R-12. Another alternative, which requires assistance from NASA, is to obtain a surplus metal bellows tank such as the ones used on the Apollo Reaction Control System Propellant Service cart, and to modify it as required.

A procurement specification has been written for the compressor, and has been forwarded to NASA-JSC. This compressor has lower performance than is ultimately possible, since it is a de-rated 5-ton compressor. This is unfortunate, however it should be satisfactory for the flight prototype testing.

The inverter for providing speed control for the compressor has not been pursued vigorously at this point. The type of unit will be similar to that using Darlington Amplifiers in Reference [4]. A potential supplier is Unitron in Garland, Texas.

2.3 Swivel Tests

The test plan for the swivel tests is being prepared at present, and is expected to be released by 29 January. The swivels are scheduled to arrive at VSD on 28 January and build-up of test equipment for the swivel test is expected to start on schedule, around 30 January. The test should start in the second week

TABLE I

SHRM COMPONENTS PROCUREMENT ACTIVITY

COMPONENT	DESCRIPTION	POTENTIAL MANUFACTURER	APPROX. COST($)	LEAD TIME (ARO)	COMMENTS
Fluid Swivel	5/8"Tube Size;Teflon Lip Seal	Fluidtronics	300 (3 pcs)	4 wk	AVO to Matls 12/19/73 - expect delivery on 1/21/74
Compressor	Heli-rotor, 3 ton 3Ø 400Hz motor	Fairchild-Stratos	5000	3 wk	6 units currently in stock at vendor-used in military ground support. Spec sent to NASA-JSC for action
Accumulator	Bellows type, 675 cu.in., 200 psid on nested bellows	Fansteel	9400	8 wk	Made up from existing tank domes, bellows from Gardner Bellows Co.
		Metal Bellows Co.	7500	14 wk	9" dia cylinder, flat ends, welded bellows
Pump/Motor	Centrifugal, 30 psi rise, 6 gpm, fluid cooled motor 3Ø 400Hz motor	Sundstrand/Pesco		19 wk	Anticipate estimate 1-21-74
		Hydro-Aire			Anticipate estimate 1-21-74. Recommend not purchasing pump at this time; use lab pump in breadboard test; shuttle pump could be used in Flight Prototype Test if dev. units are available.
By-Pass Valve	ΔP 2 psi	AiResearch	4200	8 wk	Recommend using similar valve from shuttle representative heat rejection system program
Contact HX	Q = 50,000 B/H, comprised of 20 C/P's, each 15" x 5" x 0.14" thick	UAP	10000	8 wk	Recommend dev. test program with less expensive elements to verify design
Expansion Valve	Constant super-heat; commercial type		100	1 wk	Order after compressor order is firmly placed
Inverter	Supply proper voltage and frequency to com-	Unitron			No contact has yet been made with vendor.

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of February.

3.0 CURRENT PROBLEM AREAS

There are no current technical problem areas. There are indications that component lead times of up to 14 weeks could cause a slip of 3 months or so in the breadboard test schedule. Alternative plans are being considered to avoid this slip if possible. This activity increases the length of the potential slide, so a firm decision will have to be made soon. No budgetary problem is anticipated for the length of test schedule delay that is currently projected.

4.0 WORK PLANNED FOR NEXT MONTH

The primary effort in the next month will be directed toward

(1) completion of the swivel tests on schedule, and

(2) placing orders for all critical components so that a definitive schedule can be projected for the breadboard test.

5.0 REFERENCES

[1] "Self-Contained Heat Rejection Module (SHRM) Preliminary Design Review", Enclosure to VSD Report T211-RP-004, dated 7 December 1973.

[2] Mark, H. and J. F. Cassidy, "Thermal Contact Resistance Measurements at Ambient Pressures of One Atmosphere to 3 x 10^{-12} mmHg and Comparison with Theoretical Predictions", AIAA Paper No. 69-629, AIAA 4th Thermophysics Conference, San Francisco, 16-18 June 1969.

[3] McKinzie, Jr., D. J., "Experimental Confirmation of Cyclic Thermal Joint Conductance", AIAA Paper No. 70-853, AIAA 5th Thermophysics Conference, 29 June - 1 July 1970.

[4] Wood, P, "New Contender for Motor Control", Machine Design, 10 January 1974, p. 104-106.

APPENDIX F

DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE

CONTRACT NAS9-13533
REPORT NO. T211-RP-007

PROGRESS REPORT FOR PERIOD
22 JANUARY THROUGH 7 MARCH 1974

8 MARCH 1974

SUBMITTED BY

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION
DALLAS, TEXAS

TO

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
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APPROVED BY:

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1.0 INTRODUCTION AND SUMMARY

This sixth monthly progress report under Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 22 January through 1 March 1974.

The effort during the past month was directed toward procurement of equipment for the breadboard test under the guidelines established in Reference [1], and planning of the fluid swivel test. Coldplates were obtained for use in the contact HX element test which is currently being planned.

Work planned for next month includes completion of the swivel tests and procurement activity, and release of the contact HX element Test Request.

2.0 DISCUSSION OF WORK PERFORMED THIS MONTH

2.1 Swivel Test

A test request (Reference [2]) has been released for the fluid swivel test. NASA-JSC has requested that this test be modified to include flowing relatively hot fluid past the swivel after cold soak to determine if thermal shock will damage the teflon lip seals, and if leakage will occur under this condition.

The swivel is shown in cross-section in Figure 1. In operation, the body of the swivel can rotate around the shaft which contains the inlet flow of fluid. The two lip seals prevent leakage of fluid between the shaft and the swivel body. The seals are in very close contact with metal on all sides, and have stagnant fluid on one edge. The only way the lip seal can be exposed directly to the flowing fluid temperature during transient conditions is in event of a seal leak.

VSD did not think that the lip seal will ever be subjected to thermal shock conditions (i.e., temperature changes of 100°F/sec or more) because: (1) Analysis indicates that a step temperature change of 400°F for liquid flowing into an insulated 5/8 inch ID tube, 15 feet upstream of the swivel will produce a temperature ramp of 11°F/s at the swivel. (This analysis was based on reference [3]). (2) The Modular Radiator System test conducted jointly by VSD and NASA-JSC in Chamber A at NASA-JSC involved test sequences in which it was desirable to put the maximum possible step change in fluid

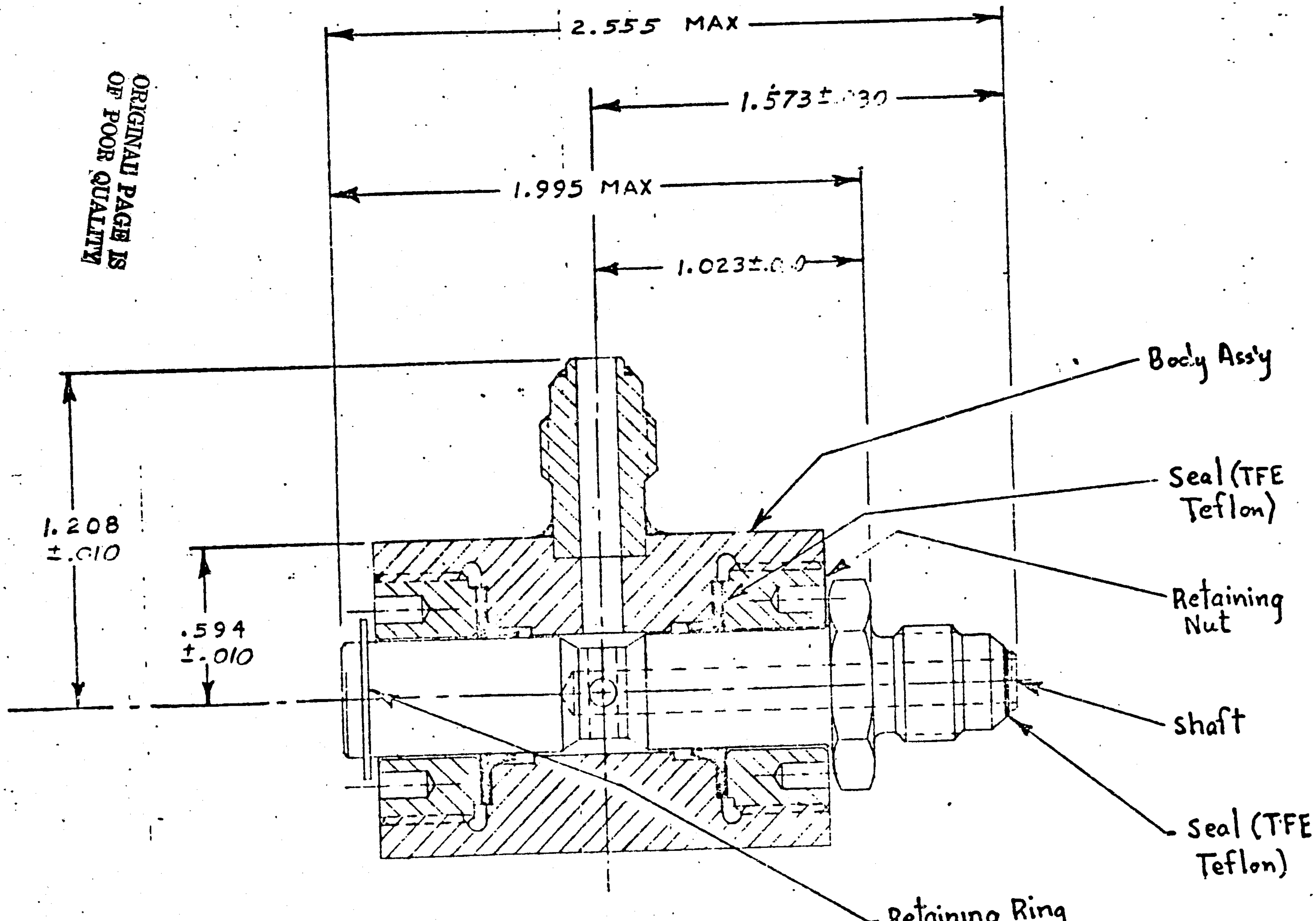

Fig. 1 FLUIDTRONICS, INC. SWIVEL

temperature on radiator panels. The maximum step change in fluid temperature which could be obtained at the radiator panels was 6°F/min. This was a very large flow system, but it seems unlikely that any practical radiator system will be able to produce step changes of more than an order of magnitude greater than this. (3) The lip seal is only 5/8-inch in I.D., 0.25-inch wide and 0.040-inch thick, which means it is very difficult to subject the seal to thermal shock while it is installed in the swivel, even with a large step in the fluid inlet temperature.

The characteristics of TFE Teflon are such that it would not be susceptible to thermal shock over the temperature range of -200°F to 200°F:

(1) TFE Teflon properties do not indicate that any molecular or crystalline structure changes will occur between -200°F and 200°F.

(2) Test conducted at VSD in which FEP Teflon film has been taken from an oven at 100°F and plunged into liquid nitrogen at -320°F have not indicated any thermal shock problems.

In addition to the above, the condition which will provide the greatest change in lip seal temperature is a failure mode in which fluid leaks past the seal, so that the fluid in contact with the seal is no longer stagnant. Even under this condition it seems unlikely that temperature changes of 100°F/sec could be expected.

It would be well beyond the scope of the test in Reference [2] to test this condition. It would require that the lip seal be modified so that it would leak prior to this test sequence. Success or failure of this part of the test would have to be based on a detailed post-test laboratory examination of the lip seal, which would be expensive. Instrumentation for this portion of the test would be expensive, and would be necessary, otherwise test conditions would not be precisely known, and thus could not be duplicated later in event of a suspected failure.

For these reasons, VSD will not conduct the requested test without NASA-JSC direction.

The condition most likely to result in failure involves flexure of the swivel at a temperature of -200°F. According to Reference [4], the

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tolerance for a lip seal on a rotating shaft less than 4-inch diameter is ± 0.003-inch. Based on a Teflon TFE expansion coefficient of 7.5 x 10^{-5} in/in-°F, a relative diameter change of 0.013-in between the seal and the shaft is expected. This is somewhat greater than the desired level; however there is reason to believe that the seal may perform adequately due to the low rotation speed of the swivel, about 0.5 rpm, and the very low number of cycles which must be endured.

The TFE fluorocarbon material was selected for the seal material because it is compatible with Freons, it operates well over a wide range of temperatures (from -320°F to 500°F), and it has good low speed properties as a seal. Its primary disadvantage lies in the fact that care must be taken during installation.

2.2 Contact HX

Two flight qualified cold plates (from the Apollo Lunar Module) have been obtained for use in the contact HX element test. Figure 2 shows the dimensions and fin characteristics of these coldplates.

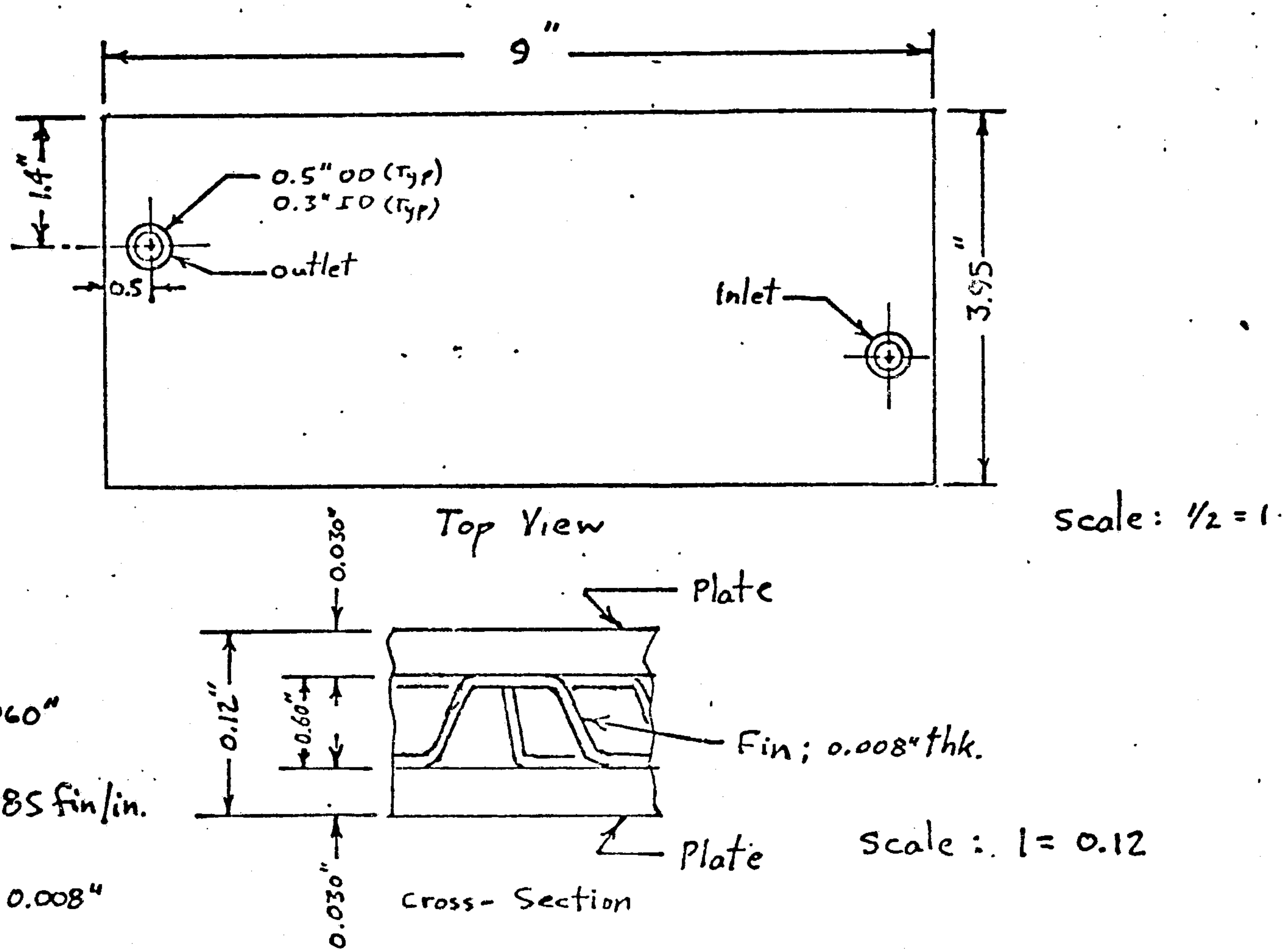
The purpose of the element test is to determine the contact conductance values which can be obtained with conventional coldplates, and to determine the variation in contact coefficient with contact pressure for these coldplates. In addition the use of interstitial foils and greases will be investigated. The proposed test set-up is shown in Figure 3.

The results of the element tests will give an indication of the practicality of the contact HX, and will guide the design of a contact HX for use in the SHRM systems test.

2.3 Procurement Activity

Brief specifications have been prepared for the refrigeration motor-compressor, the liquid pump, the accumulator, and the inverter. These are presented in Table 1 through 4, respectively.

Orders have been placed for two compressors (a heli-rotor model and a rotary model), and will be placed for other components in the near future.



Matl: Aluminum

Fin Height = 0.060"

Fin Pitch : 6.85 fin/in.

Fin Thickness = 0.008"

Fig 2 AVCO COLDPLATE DESCRIPTION

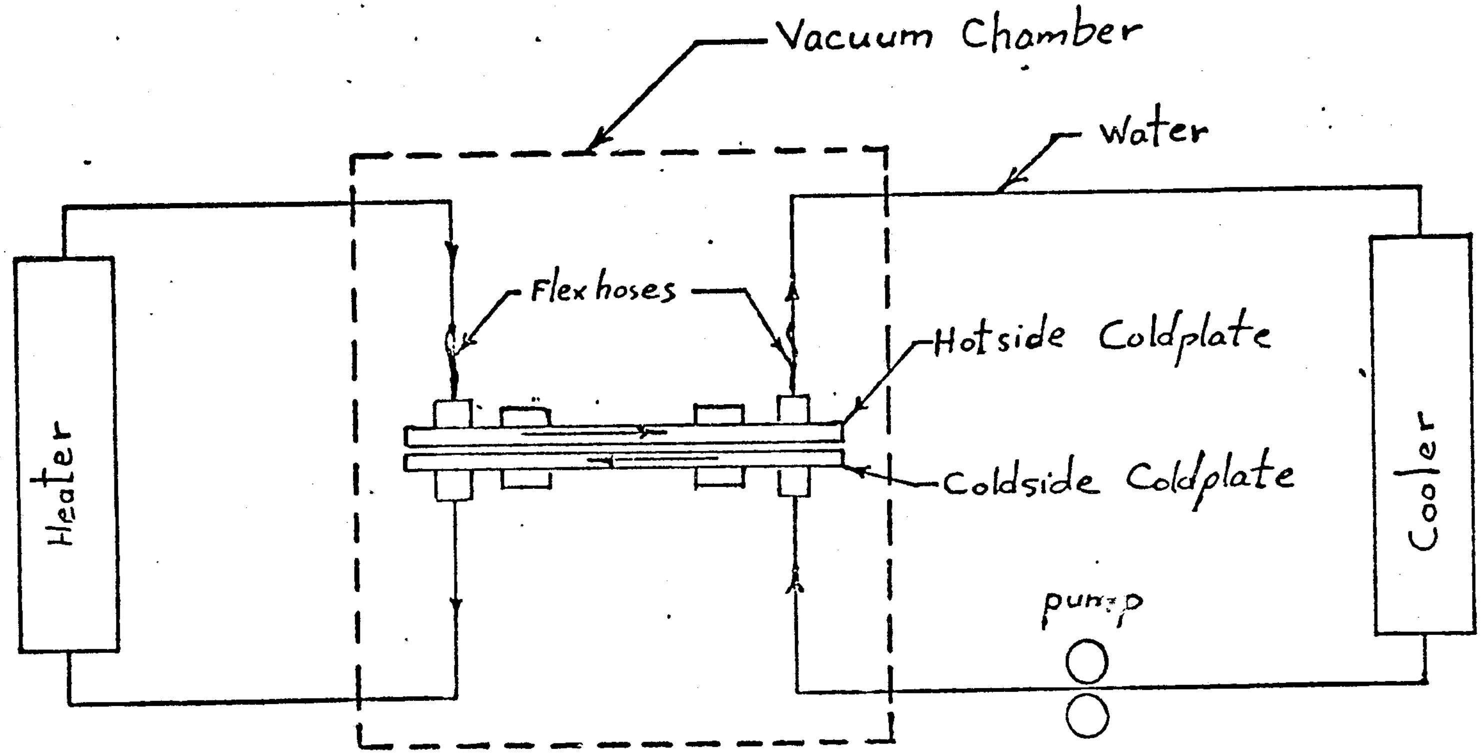


Fig 3 CONTACT HX TEST SET-UP

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TABLE I

MOTOR COMPRESSOR SPECIFICATION

DESCRIPTION:

1. Working Fluid; R12
2. Type; Heli-rotor or rotary, hermetically sealed
3. Capacity; nominal 3 tons with 120°F condenser and 40°F evaporator
4. Operation; continuous duty
5. Cooling; self-cooled by circulating refrigerant
6. Input power; 3Ø, 400 Hz, 207 volt, unit must be capable of operation at lower frequencies and voltages to provide speed control
7. Life; 2000 hours
8. Operating environment; vacuum
9. Leakage; no detectable external leakage
10. Efficiency; a COP of 2.5 at nominal operating conditions is desired

SUPPLIERS:

Fairchild-Stratos P/N HRC-5410-26
Fairchild-Stratos Division
1800 Rosecrans Ave.
Manhattan Beach, California 90266
Contact: Jack Wiseman

Task, Inc. P/N 13430-9
1009 E. Vermount St.
Anaheim, California 92803
Contact: Steve Eric

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TABLE 2

ACCUMULATOR SPECIFICATION

DESCRIPTION:

1. Working fluid: freon 12 (shell side)
   N2 (inside bellows)
2. Pressures: shell - 400 psi operating, 600 psi proof
   (equal pressure both sides of bellows)
   ΔP on nested bellows - 100 psi
3. Ports: 2 req'd, 1 to inside of bellows, 1 to shell
4. Volume: 675 cu.in. useable (minimum)
5. Configuration: cylindrical
6. Cycle life: 1000 cycles
7. Quantity Gaging: unit to include a system to establish bellows position using customer furnished lab instrumentation. A variable resistor (dashpot) connected to the moving bellows end will satisfy this requirement, if it is capable of accuracy and repeatability of within 3%.
8. Operating environment: from 14.7 ambient pressure to hard vacuum
9. Leakage: no detectable leakage either external or across the bellows

SUPPLIER:

Metal Bellows Corporation
20977 Knapp St.
Chatsworth, L.A., California 91311
Contact: Dick Roberts

TABLE 3

MOTOR/PUMP SPECIFICATION

DESCRIPTION:

1. Working fluid: R-12
2. Type: Centrifugal pump, hermetically sealed
3. Capacity: 6 gpm @ 30 psi pressure rise
4. Operation: Continuous duty
5. Cooling: Self-cooled by circulating fluid
6. Input power: 3Ø, 400 Hz, 207 volt, 400 watts max. @ .62 P.F.
7. Life: 3000 hours, minimum
8. Operating environment: Sea level ambient to hard vacuum
9. Leakage: no detectable external leakage
10. Sundstrand Model 115146-100/Modified (Ref. Sundstrand proposal No. M426-E5374-Q1 and revision 1 thereto)

SUPPLIER:

Sundstrand Aviation
4747 Harrison Ave.
Rockford, Ill. 61101
Contact: Robert P. McPhee

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TABLE 4

INVERTER SPECIFICATION

DESCRIPTION:

1. Type: solid state
2. Input power: 28 vdc
3. Output power: variable frequency from approximately 30 Hz to 400 Hz; voltage to vary linearly with frequency from 16v to 210v. The output waveform is to simulate a sinewave with at least a three square wave approximation. The output frequency should be controllable with an external signal.
4. Capacity: 6 kw
5. Service: continuous duty
6. Environment: vacuum (10^{-6} torr,
7. Life; 8000 hours
8. Efficiency: 90%
9. Weight and volume: no limitations
10. Cooling: circulating R12 at 40°F

SUPPLIER:

Not yet selected

Ths fluid swivels, which were scheduled to arrive at VSD on 28 January, have not been received. The manufacturer claims to have shipped them, so they should be received soon.

3.0 CURRENT PROBLEM AREAS

There are no technical problem areas. The swivel test schedule has been slipped due to the swivels not arriving on schedule. This test is currently scheduled for the last two weeks in March. The breadboard test has slipped approximately 3 months; a more definitive schedule will be prepared after hardware delivery dates are firmly established.

4.0 WORK PLANNED FOR NEXT MONTH

The work planned for next month includes:

(1) completion of the swivel test

(2) release of the contact HX test request

(3) initiation of preparation of drawings for the breadboard system

5.0 REFERENCES

[1] "Self-Contained Heat Rejection Module (SHRM) Preliminary Design Review", Enclosure to VSD Report T211-RP-004, dated 7 December 1973.

[2] VSD Test Request No. T211-TR-006, "Fluid Swivel Fitting Test for SHRM", dated 6 February 1974.

[3] Rizika, J. W., "Thermal Lags in Flowing Systems Containing Heat Capacitors", Trans. of ASME, April, 1954. pp. 411-420.

[4] "Dynamic Shaft Seals", Machine Design, September 13, 1973, pp.20-24.

[5] "Plastics/Elastomers Reference Issue", Machine Design, February 15, 1974, pp 16-18.

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APPENDIX G

DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE

CONTRACT NAS9-13533
REPORT NO. T211-RP-008

PROGRESS REPORT FOR PERIOD
8 March through 8 July 1974

10 July 1974

SUBMITTED BY

VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION
DALLAS, TEXAS

TO

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
HOUSTON, TEXAS

PREPARED BY:

J. L. Williams

J. L. Williams

APPROVED BY:

R. J. French

R. J. French, Supervisor
EC/LS GROUP

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1.0 INTRODUCTION AND SUMMARY

This seventh monthly progress report under Contract NAS9-13533 summarizes Vought Systems Division (VSD) activity during the period of 8 March through 8 July 1974. The effort during this period has been directed toward component tests on the fluid swivel and the contact heat exchanger (HX).

2.0 DISCUSSION OF WORK PERFORMED

2.1 Swivel Test

The three swivels obtained for the swivel component test (Reference (1)) were found to leak when originally recieved from the vendor on 15 March 1974. They were returned to the vendor for repair, and were subsequently recieved the second time on 17 June 1974. The swivels were subjected to a low temperature atmospheric leak test and one of them was found to leak.

The leaky swivel will be dis-assembled and repaired by VSD, and then the swivel component test (Reference (1)) will be conducted.

2.2 Contact HX

The contact HX element test was conducted on 24 May 1964. The Test Plan given in Appendix A: some minor changes were made in the plan to facilitate conduct of the test. The test results have been reviewed only in a cursory manner so far. The results show that the use of a thermally conductive grease between cold plates in the exchanger gives the best HX performance, and reduces the required pressure between plates to a nominal 20-30 PSI.

The results of the test will be analyzed, and a test report with recommendations for future work will be prepared in the next month.

2.3 Component Procurement Activity

The rotary compressor to be used in the laboratory prototype development test is due to be received during the week of 27 July.

1a

2.4 Laboratory Prototype Test

The Test Plan for the Laboratory Prototype test will be prepared during the next month and will be submitted to NASA for review and approval.

3.0 CURRENT PROBLEM AREAS

Current technical problem areas are:

1) The fluid swivels which seem prone to leakage. (The fluid swivel tests should provide definition of the severity of this problem).

2) Compressor Variable Frequency Power Supply (Invertor) - So far a supplier has not been identified for this component. VSD is initiating a study on the inverter.

4.0 WORK PLANNED FOR NEXT MONTH

The work planned for next month includes:

1) Completion of the fluid swivel tests

2) Release of the contact HX test report

3) Release of a draft of the laboratory prototype test plan

Figure 1 presents a revised schedule for NASA review and approval. The need dates for GFE are indicated on Figure 1.

5.0 REFERENCES

1) VSD TestRequest No. T211-TR-006, "Fluid Swivel Fitting Test for SHRM," dated 6 February 1974.

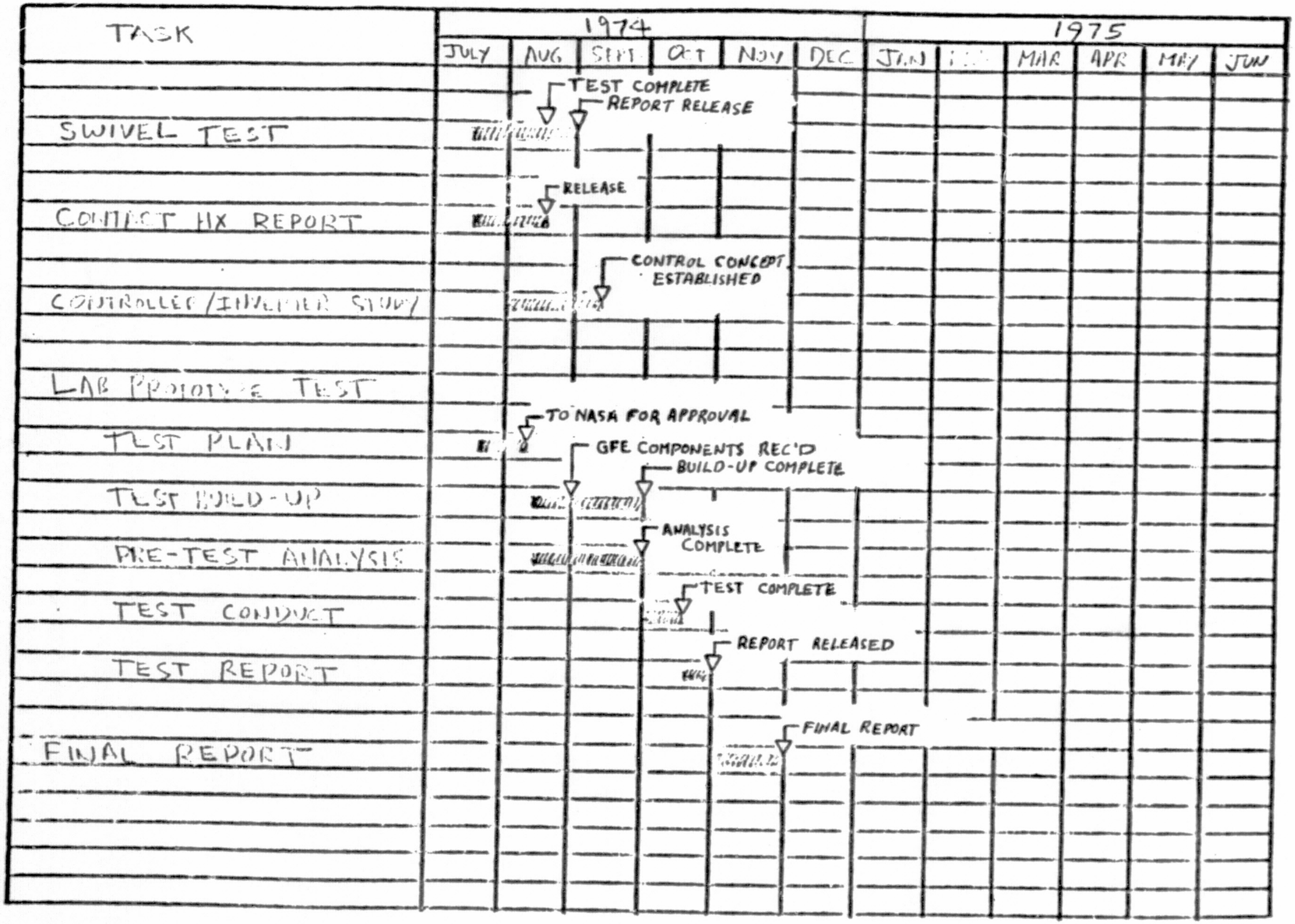

FIGURE 1 SHRM PHASE I SCHEDULE

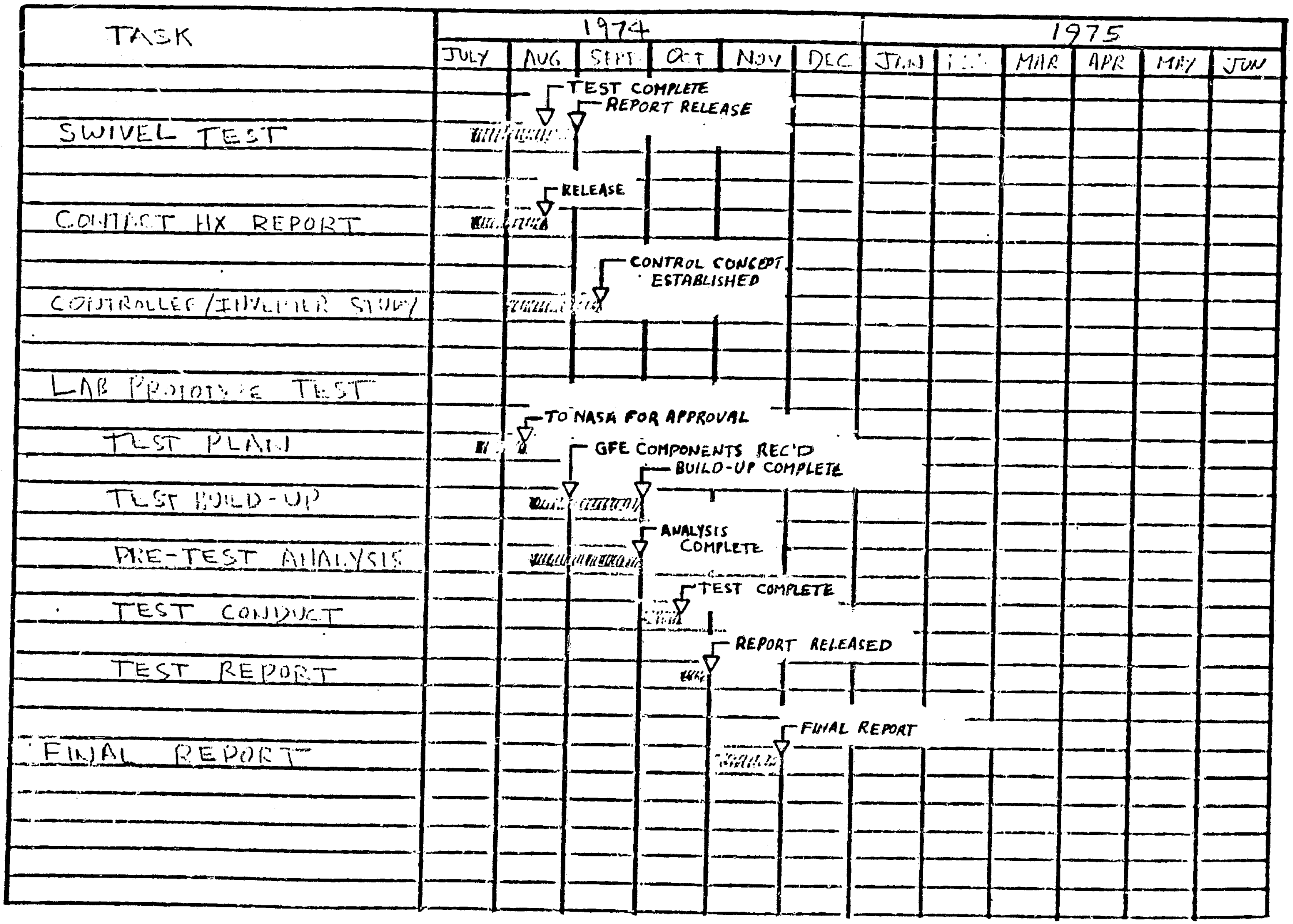

FIGURE 1 SHRM PHASE I SCHEDULE

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APPENDIX A

SHRM CONTACT HX ELEMENT TEST PLAN
Prepared 3 May 1974

TEST OBJECTIVE: Determine the heat exchanger performance of two coldplates which are pressed together. Quantify the contact heat transfer coefficients which can be obtained with various contact pressure levels, and the pressure level which is required for adequate heat exchanger performance.

TEST SETUP: The test setup is shown in Figure A-1. The coldplates are arranged so that a counterflow situation exists. The plates which apply pressure to the coldplates will have to have cut-outs for the fluid fittings as shown on Figure A-2.

INSTRUMENTATION: (1) 4 High Accuracy Immersion T/C's or Thermistors as shown on Figure A-1.

(2) 1 Flowmeter: 50-500 lb/hr range

TEST CONDITIONS: The test points desired are:

1) Interface Pressure
   30 psia
   100 psia
   300 psia
2) Flowrate
   50 lb/hr
   500 lb/hr
3) Interstitial Condition
   Bare
   DC 340 Grease
   Indium Foil
4) Inlet Temperatures
   Hot: 120 F
   Cold: 30 F
5) System Pressure: 250 psi nominal
6) Expected Pressure Drop: 2 psi in coldplates at 500 lb/hr

This represents a total of 18 test points.

TEST PROCEDURE: 1) Assemble coldplates with the selected interstitial material. (Bare first, then grease, and then foil.)
2) Pressurize the fluid loop
3) Apply the required hydraulic force
   1st - 1000 Lb_f - (~30 psi)
   2nd - 3500 Lb_f - (~100 psi)
   3rd - 10,000 Lb_f - (~300 psi)

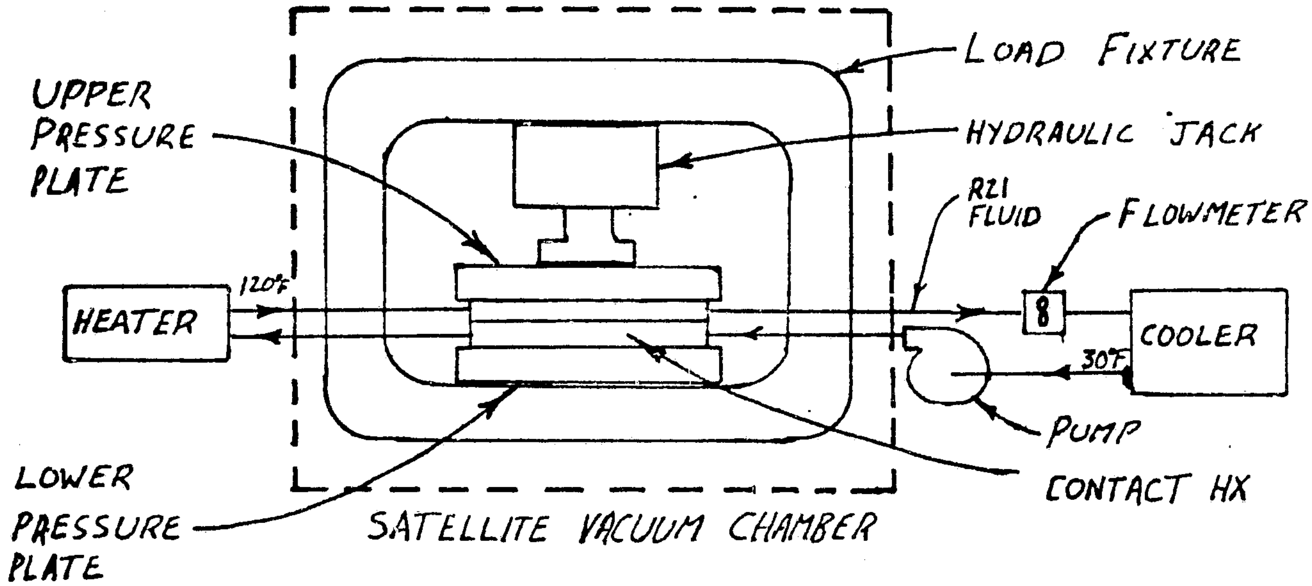

FIGURE A-1 TEST SET-UP

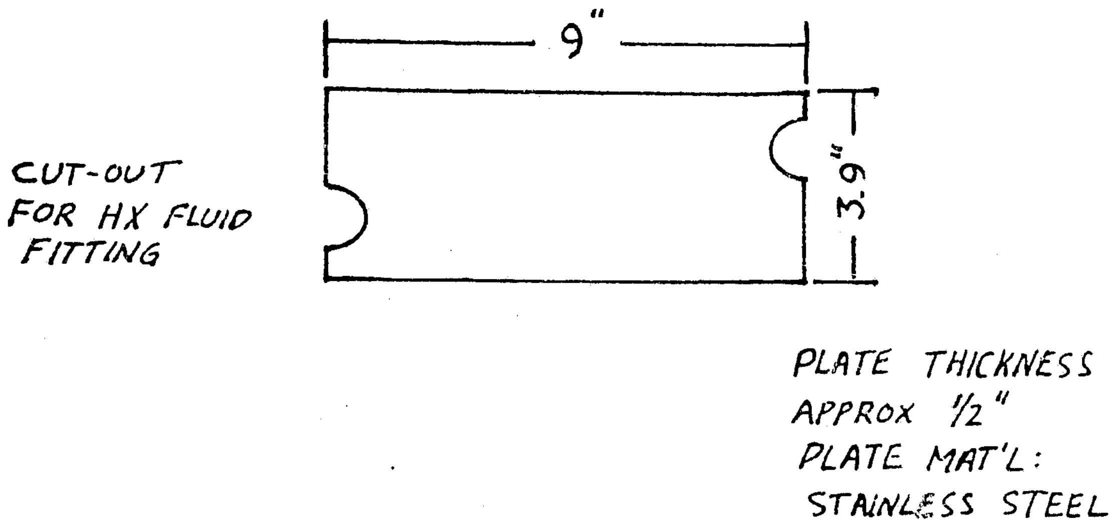

FIGURE A-2 PRESSURE PLATE

4) Establish desired Flowrate
   1st - 50 lb/hr (0.073 gpm)
   2nd - 500 lb/hr (0.73 gpm)

5) Establish steady-state conditions and record data
6) Repeat from (4) with second flowrate
7) Repeat from (3) with second and third pressures
8) Repeat from (1) with second and third interstitial conditions.

APPENDIX H

JOINT CONDUCTANCE ELEMENT TESTS FOR THE SELF CONTAINED HEAT REJECTION MODULE CONTACT HEAT EXCHANGER

Report No. T211-RP-009

15 August 1974

PREPARED BY:	REVIEWED BY:	APPROVED BY:
J. W. Leach	J. L. Williams	R. J. French
J. W. Leach	J. L. Williams	R. J. French, Supervisor EC/LS Group

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LIST OF FIGURES

LIST OF TABLES

1.0 SUMMARY

Experiments are conducted in a vacuum environment to determine the contact thermal conductance at the interface of two heat exchanger coldplates pressed together to simulate the Self Contained Heat Rejection Module (SHRM) contact heat exchanger. The element tests of this work measure joint conductance as a function of contact pressure for bare metal interfaces and for joints with lead foil and thermal grease interstitial materials. The results show that acceptable levels of the interface conductance can be achieved at relatively low contact pressures with thermal grease. Acceptable levels cannot be obtained with bare metal contact or with lead foil unless the contact pressure is high or the contact surfaces are specially prepared to reduce roughness and waviness.

2.0 INTRODUCTION

Many experiments (2-8)* have been conducted which investigate the transfer of heat across contact interfaces for the purposes of reducing or inhancing the heat transfer. Interface or contact conductance is a function of the presence or lack of an interstitial layer, contact pressure, surface roughness, and indentation hardness. When two materials are in contact heat flow is enhanced by one or more of the following: increasing the contact pressure, reducing the surface roughness, and reducing indentation hardness. All of these factors tend to increase the fraction of the interface area which comes into contact when the surfaces are pressed together. Interstitial materials with high thermal conductivities increase the joint conductance because they fill small voids at the interface caused by surface roughness and waviness.

Previous experiments have been of a basic nature and have tested relatively small coupon sized contact areas which were specially prepared to insure flatness and uniformity of surface characteristics. It is questionable as to whether data obtained from such experiments are really applicable to the SHRM contact heat exchanger because the surfaces in this case are neither uniform nor flat. The face sheets of the heat exchanger core are brazed to internal fins designed to increase the convective heat transfer. The brazing process creates large scale surface irregularities which reduce the fraction of the interface area available for heat transfer. Large amplitude waves which occur where the face sheets attach to the fins probably have a much larger effect on the joint conductance than surface roughness or indentation hardness as studied by earlier investigators.

This work is designed to determine the values of the joint conductance that are possible with heat exchanger coldplates typical of the SHRM application. Previous studies (1) have shown that contact conductance values of the order of 1000 BTU/hr-ft^2-$^\circ$F are desirable for reducing the weight and volume of the SHRM contact heat exchanger. Several interstitial materials are studied herein to determine whether acceptable values of the contact conductance can be attained with contact pressures typical of bolted joints.

*Numbers in parenthesis refer to the list of references in Section 6.0

3.0 TECHNICAL DISCUSSION

3.1 Test Objective

The objective of the SHRM joint conductance element test is to determine the value of the contact heat transfer coefficient between two coldplates which are pressed together in a vacuum environment for various contact pressure levels and interstitial filler materials.

3.2 Test Article and General Requirements

The contact conductance between two identical 3.9" x 9" coldplates was measured. The joint conductance was determined from changes in the properties of a transport fluid (Freon 21) which was flowed through the coldplates. A description of the coldplates is given in Table I and the general test requirements are given in Table II. A radiographic photograph showing the flow passages in the coldplate is given in Figure 1. The contact heat transfer coefficient was measured with bare metal contact of the coldplate and with interstitial filler materials described in Table III being placed between the contact surfaces. Data was recorded for sequences of increasing and decreasing loads to determine the effect of compressing the filler material.

3.3 Joint Conductance Calculations

The value of the joint conductance at the interface of contact heat exchangers cannot be measured directly but can be deduced from measurements of the total thermal resistance between the two fluids of the heat exchanger. In this work the overall conductance is determined by measuring the fluid inlet and outlet temperatures which are employed in the following equation.

$$q = U_o A_p \, \Delta T_M = \dot{m} \, \Delta h \qquad (1)$$

where:

- q = heat transfer from hot to cold fluid
- U_o = overall conductance
- Ap = contact surface area
- ΔT_M = log mean temperature difference
- $\dot{m}$ = flowrate of hot or cold fluid
- Δh = enthalpy change of hot or cold fluid

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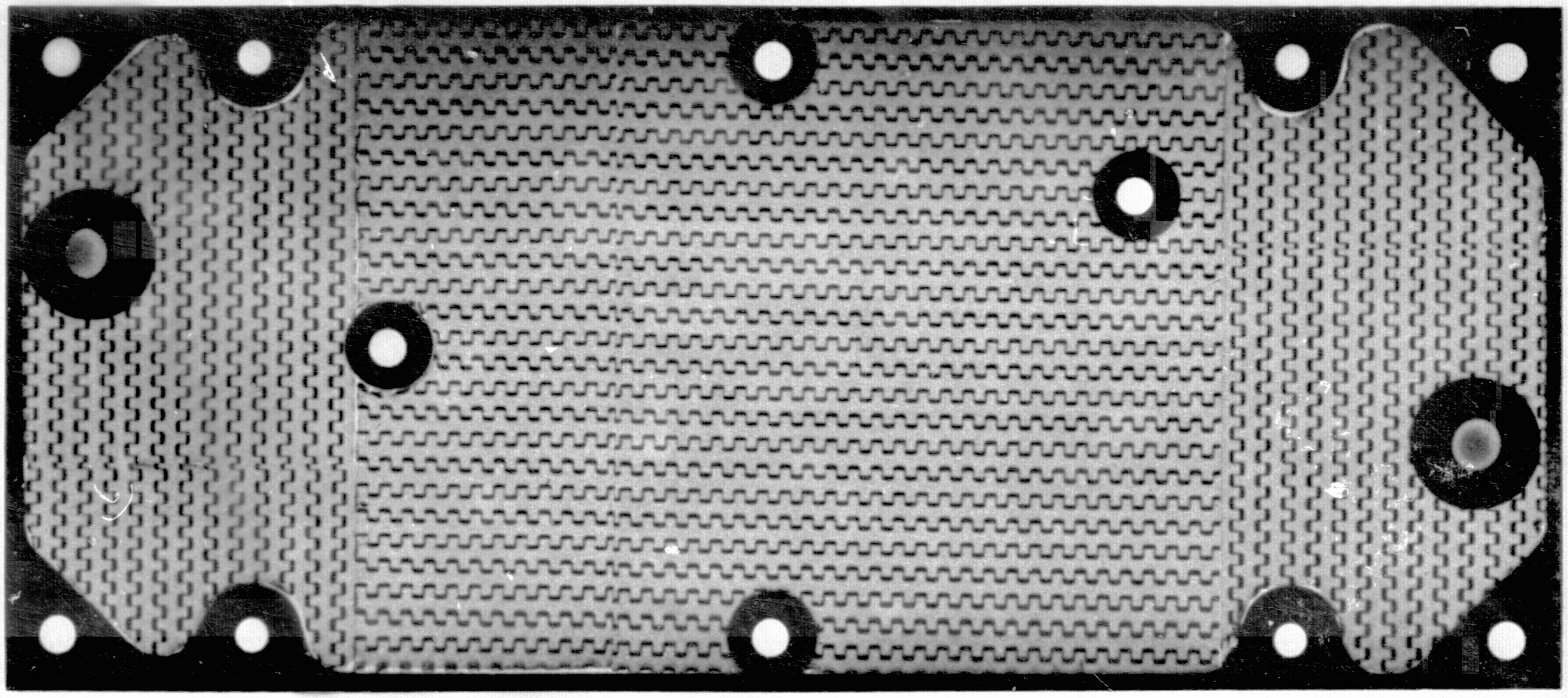


FIGURE 1 RADIOGRAPH OF COLDPLATE

TABLE I

DESCRIPTION OF TEST ARTICLE

$\leftarrow 0.243\,ft^2$

OVERALL DIMENSIONS	3.9" x 9"
SURFACE ROUGHNESS	45μin. rms
FIN TYPE	Pierced
FIN HEIGHT	0.090"
FINS/INCH	6.7
HEAT TRANS. AREA/PLATE AREA	3.12
FLOW AREA/WIDTH, EASYWAY	.00624'
FLOW AREA/WIDTH, HARDWAY	.00340'
HYDRAULIC DIAMETER, EASYWAY	0.00800'
HYDRAULIC DIAMETER, HARDWAY	0.00438'
PLATE THICKNESS	0.030"

TABLE II

GENERAL TEST REQUIREMENTS

FLUID	FREON 21
FLOWRATE	500 lb/hr
MAXIMUM CONDITIONED TEMPERATURE	120°F
MINIMUM CONDITIONED TEMPERATURE	30°F
ENVIRONMENT	Vacuum (10^{-4} torr)
CONTACT LOAD RANGE	0-10,000 lb_f
INTERFACE PRESSURE	0-300 psia
SYSTEM PRESSURE	250 psia

TABLE III

INTERSTITIAL FILLER MATERIAL PROPERTIES

MATERIAL	Thermal Conductivity (BTU/hr-ft-°F)
0.005" Thick Lead Foil	20.1
DOW CORNING DC340 GREASE	0.362
GENERAL ELECTRIC G641 GREASE	0.428

The overall conductance includes the joint conductance, convective heat transfer coefficients within the heat exchanger core, and terms which account for conduction through the fins and plates of the core.

$$\frac{1}{U_o} = \frac{1}{U} + \left(\frac{1}{\eta\,(\frac{A}{Ap})\,h}\right)_{\substack{\text{HOT}\\\text{FLUID}}} + \left(\frac{1}{\eta\,(\frac{A}{Ap})\,h}\right)_{\substack{\text{COLD}\\\text{FLUID}}} \qquad (2)$$

where: U_o = joint contact conductance

η = overall fin conduction efficiency

A = convective heat transfer area

h = convective heat transfer coefficient

The terms in Equation 2 which stem from convective and conduction heat transfer internal to the heat exchanger core are measured in separate experiments which do not involve the joint conductance. Data obtained from such experiments were provided by the core manufacturer in the form of dimensionless curves designed to remove the dependence of the core performance upon the transport fluid. The performance curves for the core of this work obtained for Freon 113 are given in Figure 2. In Appendix I the universal core performance data are combined with Freon 21 property data to give the internal heat transfer terms in Equation (2).

To minimize the effects of unknowns in Equation (2) it is desirable to make the resistances to heat transfer within the core as small as possible. This requires that the flowrates be relatively high so that turbulent flow is achieved. Also relatively large changes in the fluid temperature are desirable for minimizing experimental errors in the temperature recording system. This can be accomplished at high flowrates by regulating the size of the coldplates and the difference between the hot and cold fluid inlet temperatures. The heat exchanger area, flow rate, and fluid temperatures selected in this work give a reasonable representation of the actual flow conditions in the SHRM application while insuring that accurate joint conductance data can be computed from Equations (1) and (2).

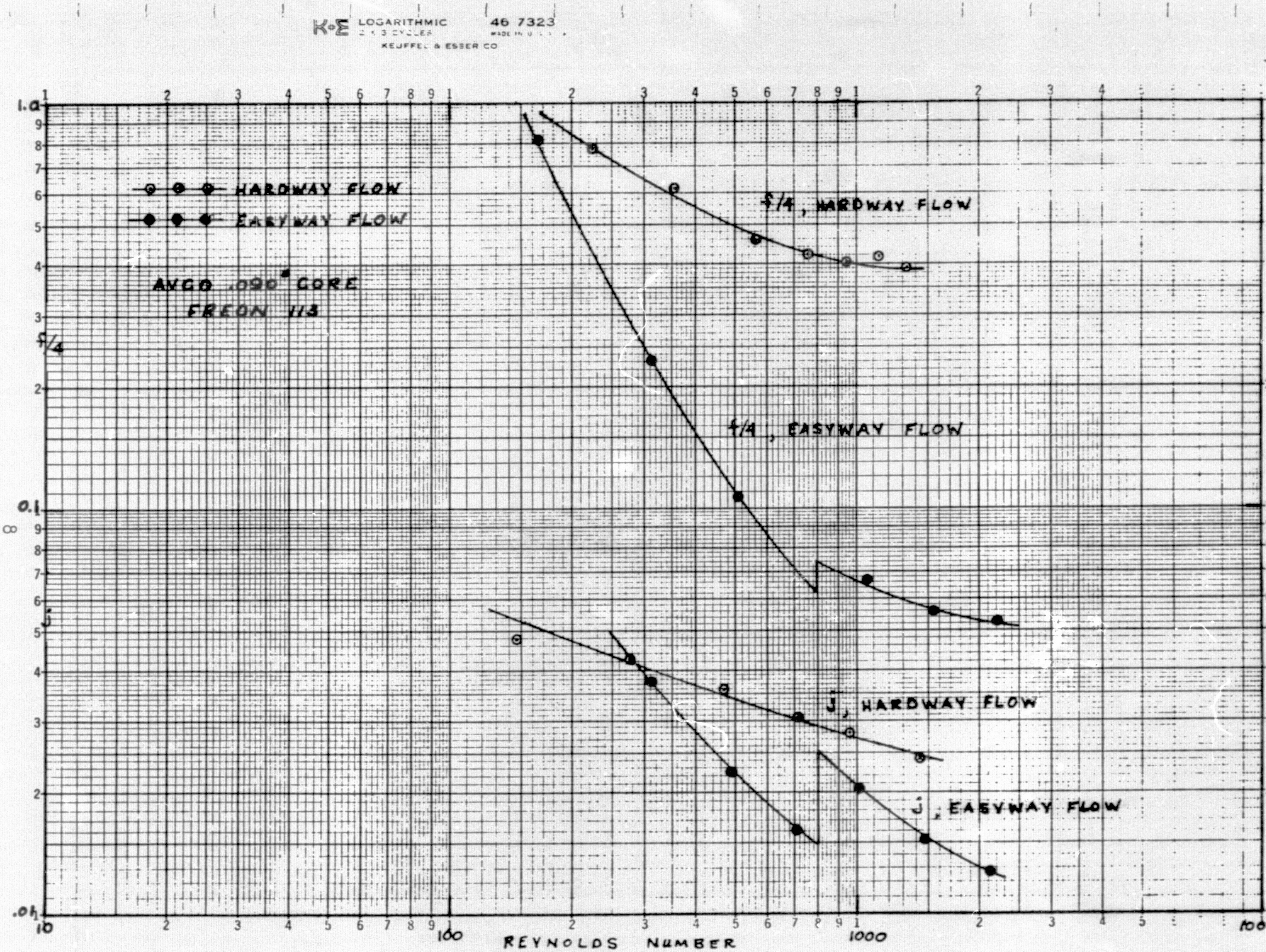


FIGURE 2 CORE PERFORMANCE DATA

3.4 Description of Apparatus

A schematic of the test apparatus is shown in Figure 3. The fluid temperatures at the coldplate inlets were controlled with a coolant fluid (Trichloroethylene) and with electrical resistance heaters. The flowrate of the transport fluid (Freon 21) was controlled and monitored using valves and turbine flowmeters located on the Freon 21 flowbench shown in Figure 4. The temperature of the fluid entering the flow bench was maintained at approximately 65°F to minimize the effects of fluid property variations with temperature on the flow measurements.

The surface contact load was applied with a hydraulic jack and measured with a ring type load cell shown in Figure 5. The apparatus was installed in the VSD satellite vacuum chamber as shown in Figure 6. The fluid temperatures entering and leaving the coldplates were measured with sheathed and unsheathed immersion copper constantan thermocouples. The thermocouple signals were recorded on a Brown strip recorder with a readability of 0.1°F. Errors introduced by the reference junction, thermocouple wire, and by heat leaks limit the accuracy of the temperature recording system to approximately 3°F. The precision of the system as established from thermocouple consistency tests is $\pm$ 0.5°F. Since the temperature differences recorded during the test range from 20°F at low contact pressures with bare metal contact to 40°F at high contact pressures with the thermal grease interstitial materials, thermocouple errors could introduce errors of the order of 5%. The experiment was designed with redundant temperature measurements so that thermocouple errors could be evaluated from differences in independent measurements. Thermocouple consistency tests and vacuum chamber test data discussed below indicate that the actual errors introduced by the temperature recording system are within this margin.

The flowmeters were calibrated with water at the end of the test and were found to be accurate within 0.5%. Fluid density variations affect the flowmeter output because the frequency of the turbine depends on the volumetric flowrate rather than the mass flow rate. Since the fluid temperature at the flow bench varied between 60°F and 70°F this could introduce an additional error in the flowmeter reading of 0.9%.

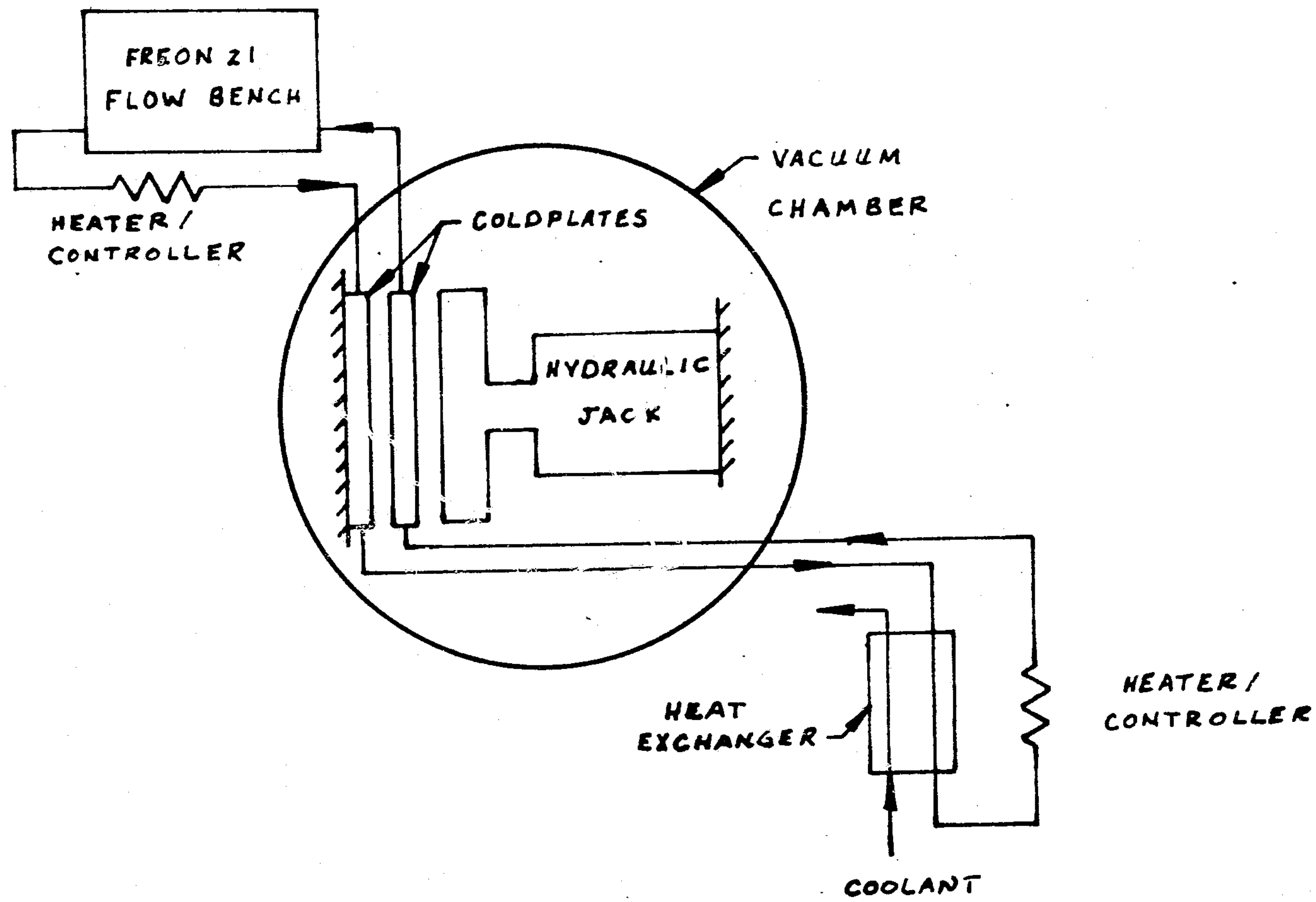


FIGURE 3 SCHEMATIC OF APPARATUS

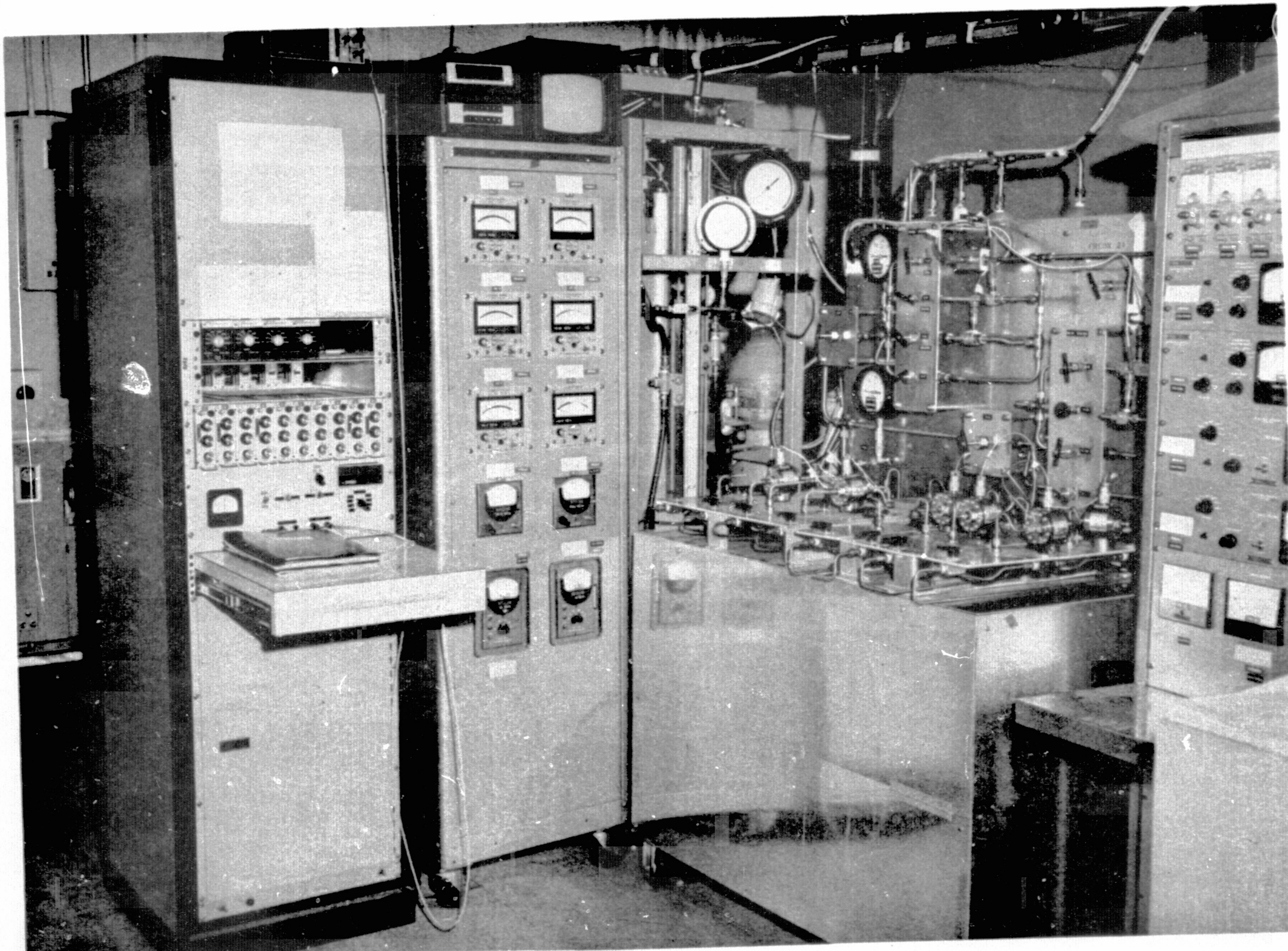


FIGURE 4 PHOTOGRAPH OF FREON 21 FLOW BENCH

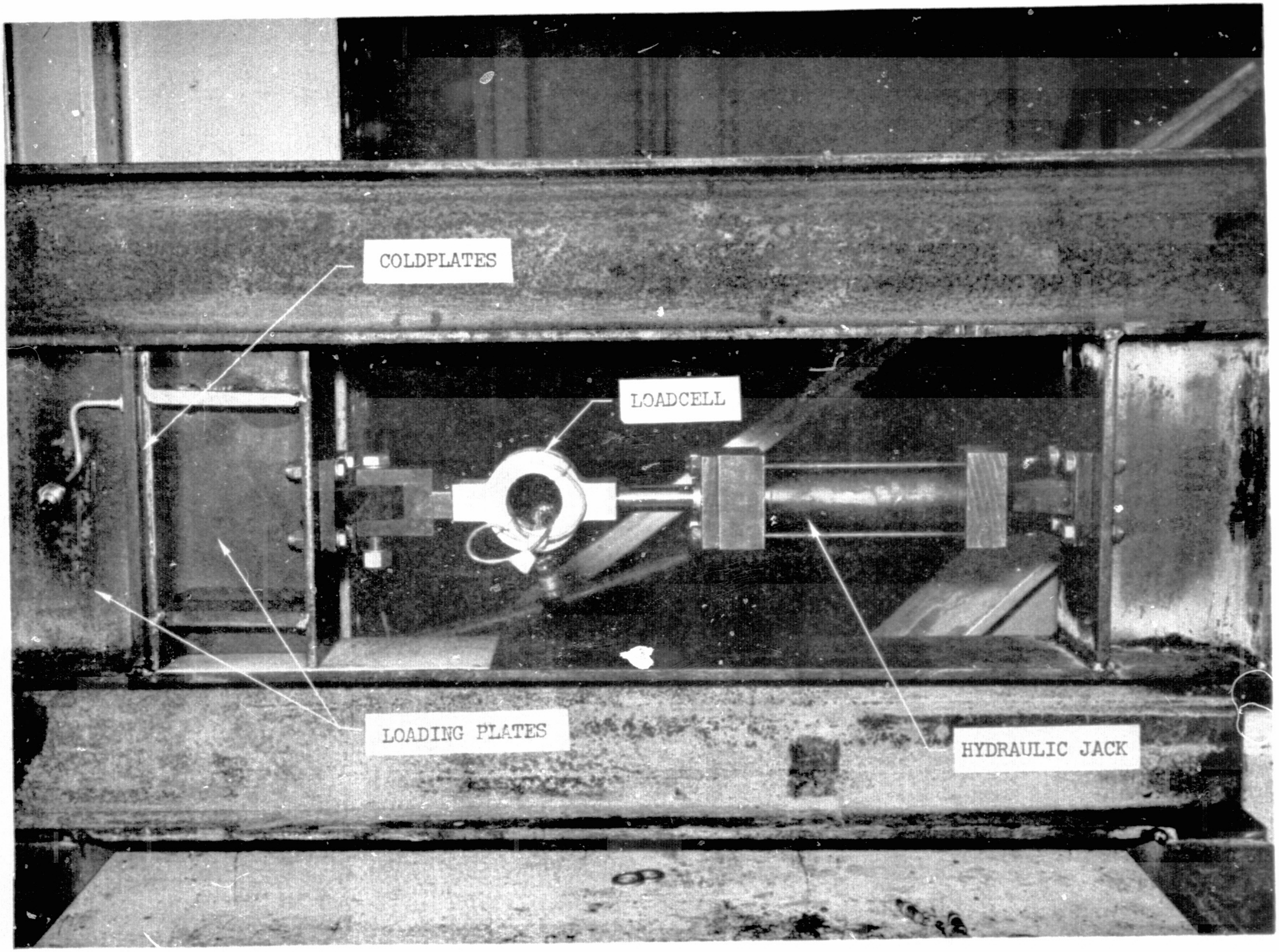


FIGURE 5 PHOTOGRAPH OF APPARATUS

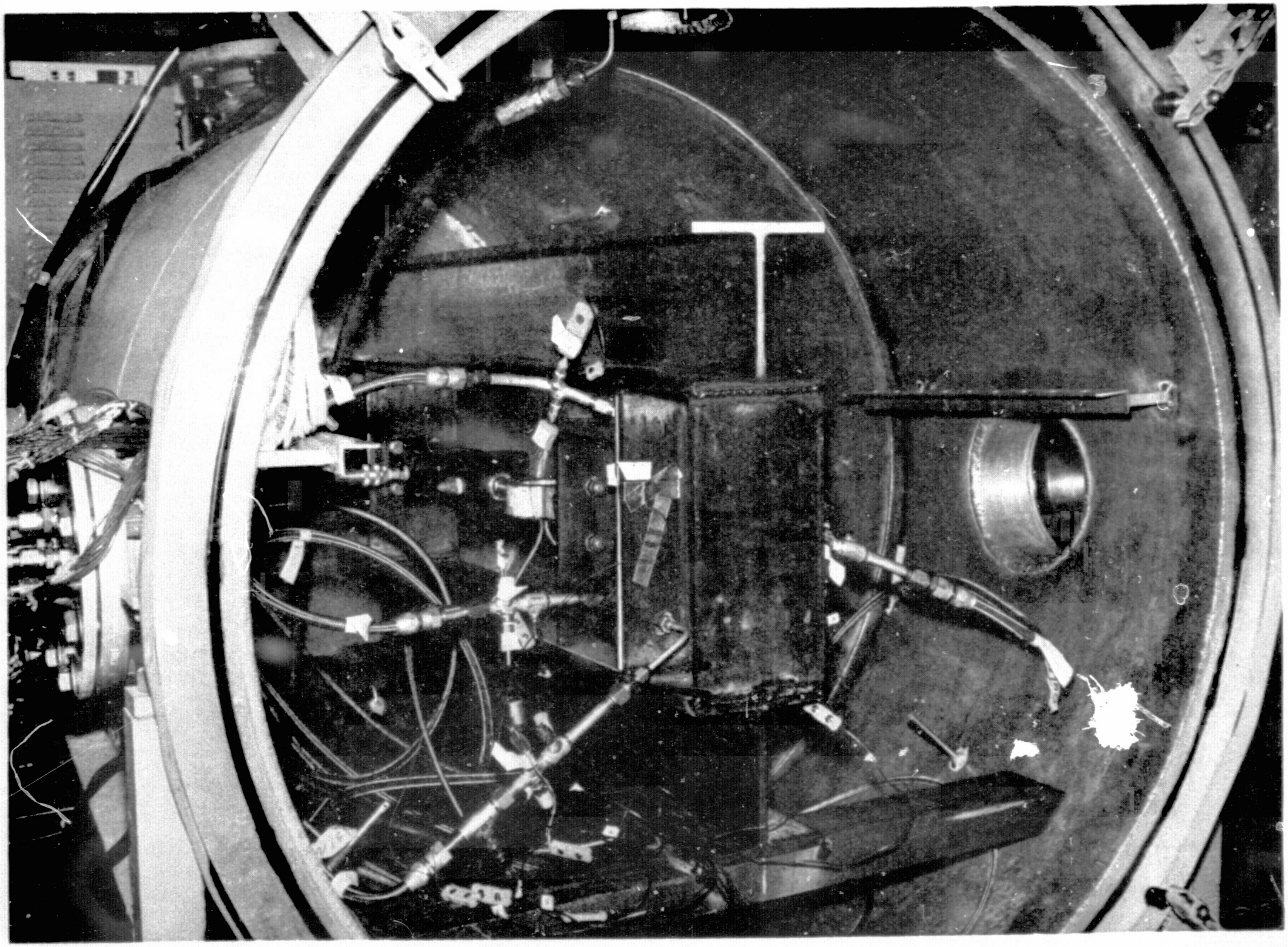


FIGURE 6 INSTALLATION OF TEST ARTICLE IN VACUUM CHAMBER

3.5 Instrumentation Consistency Tests

Tests were conducted to determine the consistency of the thermocouple readings and to evaluate the effects of heat leaks on the accuracy of the joint conductance measurements. The temperatures were recorded with no coolant being circulated through the trichloroethylene - Freon 21 heat exchanger. This caused the hot and cold fluid Freon 21 inlet temperatures to be equal so that each thermocouple would theoretically read the same. Table IV gives the fluid temperatures recorded during this test. The results show that the sheathed and unsheathed thermocouples agreed within 1°F and that heat leaks to the environment are minimal. Heat losses which occurred during the contact conductance test were evaluated from the difference in the enthalpy changes of the hot and cold fluids. This data is presented along with the sheathed and unsheathed thermocouple readings in Table V. The results show that the thermocouple readings are very consistent and substantiate that the instrumentation and thermal insulation are entirely satisfactory for determining the heat transfer in the joint conductance test.

3.6 Loading Structure

The load was applied to the coldplates through a 6" x 6" wide flange I-beam shown in Figure 5. The beam was reinforced with 0.5" thick plates which were welded to the flanges to minimize deflections on the loading surface. The face of the flange in contact with the coldplates was machined flat to give a uniform contact pressure. The backplate supporting the opposite coldplate was made from an identical section of I-beam and was welded to the supporting structure. The face of the backplate was not machined flat but the surface was checked for warpage after welding. Teflon sheets were placed between the loading plates and the coldplates to thermally insulate the coldplates from the loading structure.

3.7 Test Plan

Table VI gives the test sequence for the SHRM contact conductance element test.

TABLE IV

THERMOCOUPLE CONSISTENCY TEST DATA

RUN NO.	COLD INLET #1	COLD INLET #2	COLD OUTLET #3	HOT INLET #4	HOT OUTLET #5	HOT OUTLET #6
1	102.0	101.5	101.1	101.1	101.1	100.7

TABLE V JOINT CONDUCTANCE TEST CONSISTENCY TEST

RUN NO.	DIFFERENCE IN COLD INLET THERMOCOUPLES (°F)	DIFFERENCE IN HOT OUTLET THERMOCOUPLES (°F)	ENTHALPY CHANGE HOT FLUID (BTU/LB)	ENTHALPY CHANGE COLD FLUID (BTU/LB)	HEAT LOSS
3	+ .3	.6	5.56	5.53	.03
4	+ .6	.7	7.22	7.28	-.05
5	+ .5	.5	7.11	7.29	-.18
6	+ .2	.1	8.77	8.92	-.15
7	+ .1	.4	7.37	7.06	.31
8	0	.5	10.35	10.10	.25
9	0	.3	11.73	11.56	.17
10	0	0	12.93	12.75	.17
11	- .2	.9	1.79	1.63	.16
12	- .3	1.1	2.09	1.95	.14
13	- .1	.9	2.52	2.40	.12
14	- .2	.9	3.11	3.04	.07
15	- .2	1.0	3.98	3.88	.10
16	- .2	.8	5.51	5.43	.08
17	- .2	.5	6.78	6.69	.09
18	- .2	.5	7.83	7.76	.07
19	- .2	.4	8.95	8.89	.06
20	- .3	1.0	5.99	5.90	.09
21	- .3	1.1	4.77	4.66	.11
22	- .2	1.1	3.10	3.08	.06
23	- .3	1.0	4.46	4.40	.06
24	- .2	.5	7.40	7.39	.01
25	- .2	.5	7.78	7.73	.05
26	- .2	.4	8.26	8.20	.06
27	- .2	.3	8.62	8.57	.05
28	- .2	.2	8.93	8.89	.04
29	- .2	.5	9.15	9.06	.09
30	- .2	.2	9.21	9.14	.07
31	- .2	.1	10.10	10.08	.02
32	- .2	.1	10.02	9.98	.04
33	- .2	.2	9.18	9.16	.02
34	- .3	.3	8.67	8.62	.05
35	- .3	.8	8.11	8.00	.11
36	- .8	0	12.31	12.10	.21
37	- .2	.6	7.80	7.78	.02
39	- .2	0	10.07	10.02	.05
40	- .3	0	10.53	10.42	.11
41	- .2	.3	10.74	10.57	.17

TABLE VI

CONTACT CONDUCTANCE ELEMENT TEST SEQUENCE

RUN NO.	INTERSTITIAL MATERIAL	LOAD RANGE (lbf)	FLOWRATE (lb/hr)
3-7	Bare Metal	1000-10,000	500
8-10	Bare Metal	1000-10,000	100
11-19	Lead Foil	100-10,000 (loading)	500
20-23	Lead Foil	10,000-1000 (unloading)	500
24-31	DC-340 Grease	100-10,000 (loading)	500
32-35	DC-340 Grease	10,000-0 (unloading)	500
36	DC-340 Grease	1000	100
37-40	G-641 Grease	100-10,000	500
41	G-641 Grease	2000 (unloading)	500

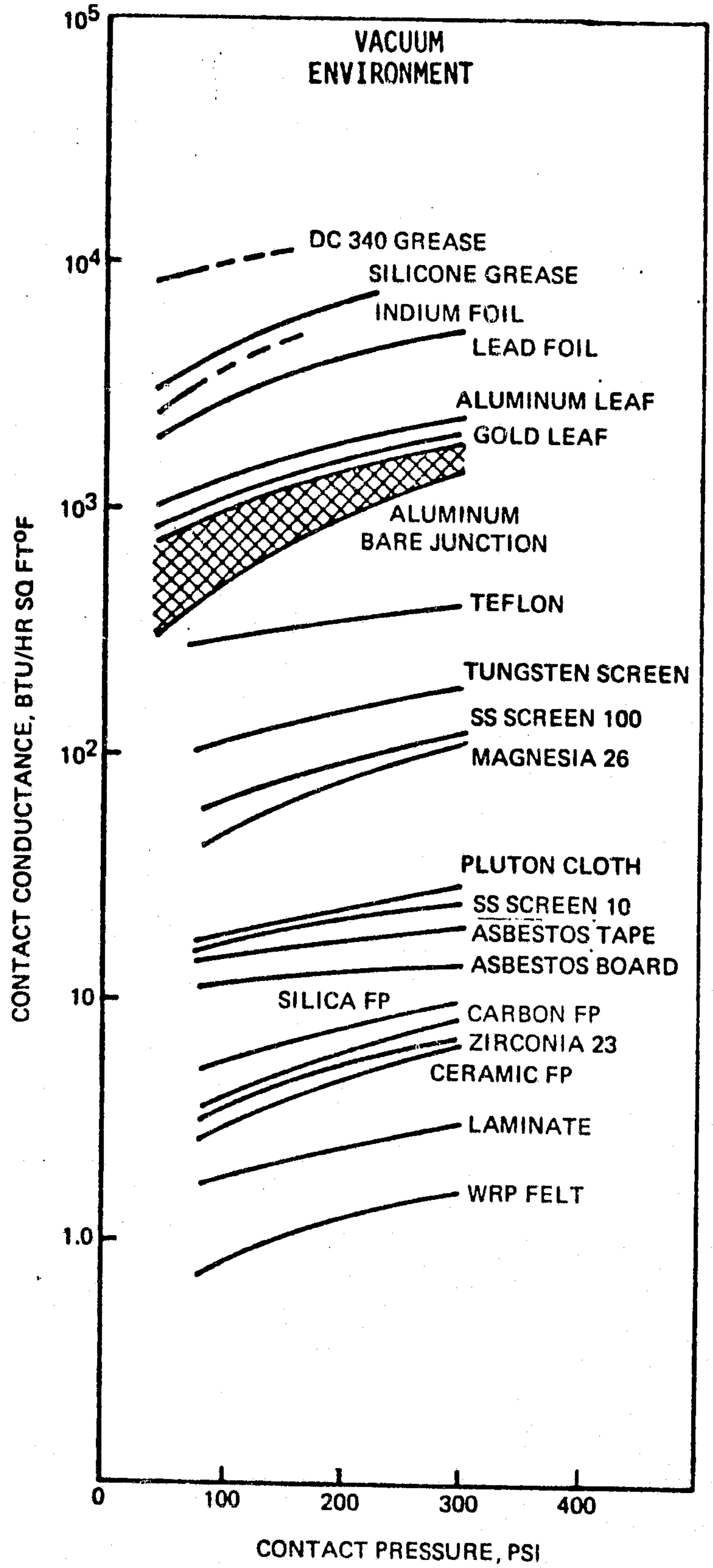


FIGURE 7 COMPARISON OF THERMAL CONDUCTANCE FOR SELECTED THERMAL CONTROL MATERIALS

4.0 RESULTS

Details of the calculations of the overall resistance to heat transfer from the experimental data are given in Table VII. The values in Table VII are combined with the internal heat transfer resistances from Appendix I to give the joint conductances in Table VIII. The calculations in Tables VII and VIII follow Equations (1) and (2) of Section 3.3.

The internal thermal resistance computed in Appendix I is subject to inaccuracies in the core performance data and assumes that the flow distribution inside the coldplate is the same as that of the core originally tested by the manufacturers. Actually the flow inside the coldplates is nonuniform because of entrance region effects and flow interference at bolt hole locations. This could cause the actual thermal resistance to differ from the predicted value. The internal resistance can be computed from the data of this work if one assumes that the joint conductance is very large for the thermal greases at high contact pressures. Data of other investigators (8) given in Figure 7 shows that a joint conductance of approximately 10,000 BTU/hr-ft^2-°F should be expected at a contact pressure of 300 psia. This yields a total internal resistance of 0.0022 Hr-Ft^2-°F/BTU which may be compared with the predicted value of 0.0016 Hr-Ft^2-°F/BTU. The agreement is as good as can be expected considering the sources of error discussed above. This difference in the internal resistance has little impact on the joint conductance computed for bare metal contacts or for the lead foil interstitial material. However, as is shown in Table VIII the joint conductances computed for the thermal greases at high contact pressures are influenced significantly by this discrepancy. Values of the joint conductance computed using both values of the internal resistance are plotted in Figure 8. The size of the shaded areas indicate the magnitude of the error introduced by uncertainties in the internal thermal resistance. Although the errors are very large in some cases, the results show that the joint conductance is sufficiently large for the SHRM applications. Previous analyses showed that joint conductance values of 1000 BTU/hr-ft^2-°F are desirable for minimizing the weight and volume of the SHRM contact heat exchanger. This is easily achieved with either of the thermal greases studied in this work.

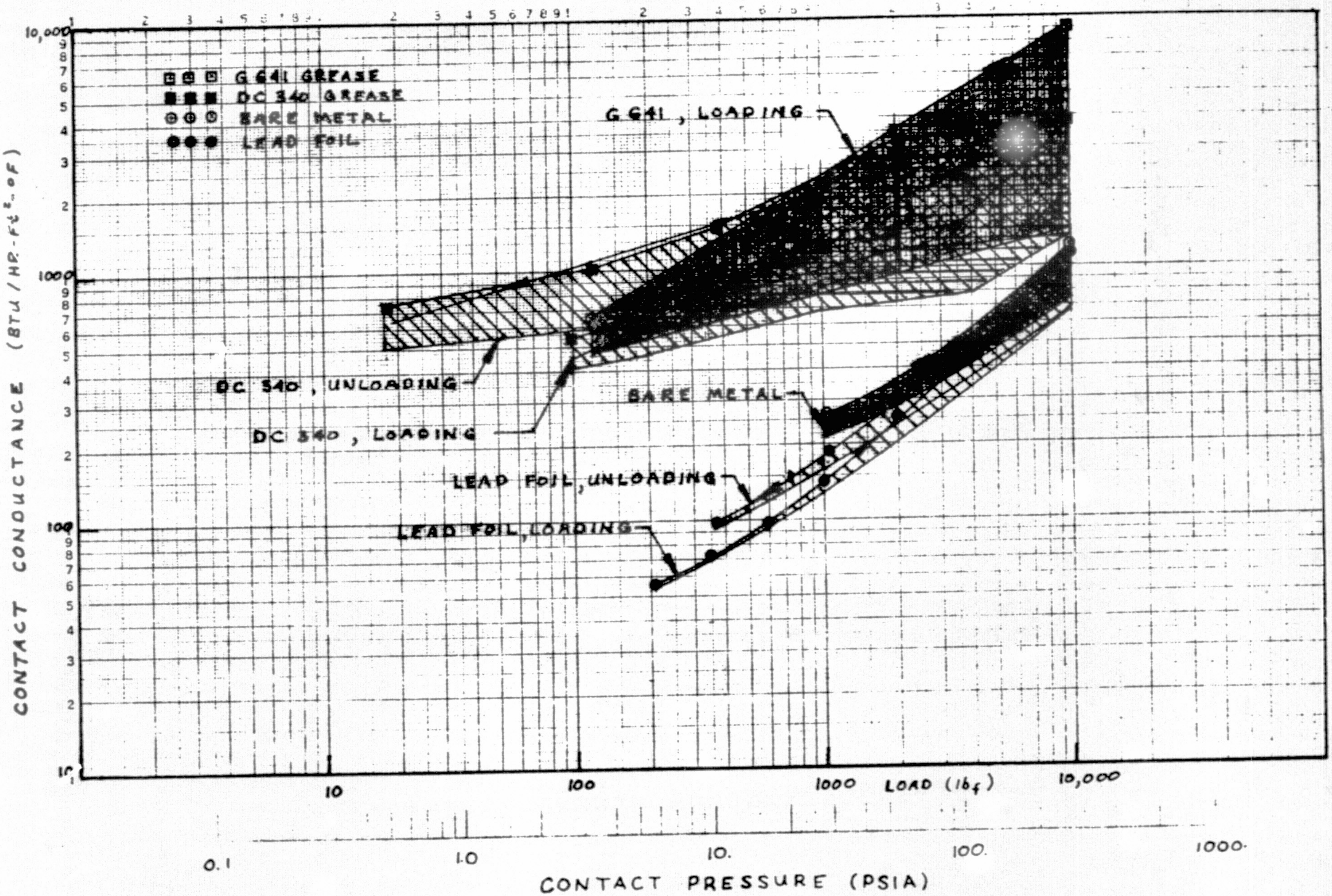
Figure 9 from Reference (2) shows the effect of surface roughness on the contact conductance of bare aluminum. The results of this test fall below

TABLE VII DETAILS OF OVERALL CONDUCTANCE CALCULATIONS

RUN NO.	LOAD (lb_f)	$\dot{w}$ (lb/hr)	$T_{HOT\ IN}$ (°F)	$T_{HOT\ OUT}$ (°F)	$T_{COLD\ IN}$ (°F)	$T_{COLD\ OUT}$ (°F)	Δh (BTU/lb)	ΔT_m (°F)	1/Uo
3	1019	500	119.7	97.7	52.2	29.8	5.55	67.6	.00597
4	3500	500	119.6	91.3	59.0	29.5	7.25	61.2	.00414
5	3500	500	119.7	91.8	59.0	29.5	7.20	61.7	.00420
6	10000	500	120.2	85.8	66.2	30.1	8.85	54.8	.00303
7	3500	500	120.9	92.0	59.0	30.4	7.21	61.8	.00420
8	1015	100	118.8	78.2	74.5	33.6	10.22	44.5	.01067
9	3500	100	118.9	72.9	80.0	33.2	11.64	39.3	.00828
10	10,056	100	119.0	68.3	84.9	33.3	12.84	34.4	.00657
11	110	500	120.6	113.6	37.4	30.8	1.71	33.0	.00945
12	210	500	120.9	112.7	38.9	31.0	2.02	81.9	.01987
13	350	500	121.1	111.2	40.8	31.1	2.46	80.2	.01597
14	600	500	121.4	109.2	43.5	31.2	3.07	78.0	.01245
15	1005	500	121.5	105.9	46.8	31.1	3.93	74.7	.00931
16	1981	500	121.9	100.3	53.1	31.1	5.47	69.0	.00618
17	3500	500	122.3	95.7	58.3	31.2	6.73	64.2	.00468
18	6000	500	122.6	91.9	62.6	31.2	7.79	60.3	.00379
19	10,000	500	122.9	87.8	67.2	31.2	8.92	56.1	.00308
20	2020	500	122.5	99.0	55.2	31.3	5.94	67.5	.00557
21	1057	500	122.3	103.6	50.2	31.3	4.71	72.2	.00751
22	370	500	122.0	109.7	43.8	31.3	3.11	78.3	.01233
23	1000	500	122.3	104.8	49.2	31.4	4.43	73.2	.00810
24	98	500	121.5	92.5	61.1	31.2	7.39	60.9	.00404
25	203	500	121.5	91.0	62.5	31.2	7.75	59.4	.00376
26	363	500	121.7	89.3	64.4	31.2	8.23	57.7	.00344
27	594	500	121.8	88.0	65.9	31.2	8.59	56.4	.00322
28	1000	500	121.9	86.9	67.2	31.2	8.91	55.2	.00304
29	2000	500	122.0	86.1	67.9	31.2	9.11	54.5	.00293
30	3500	500	122.0	85.9	68.2	31.2	9.17	54.2	.00290
31	10000	500	122.0	82.4	72.0	31.2	10.09	50.6	.00246
32	2106	500	122.2	82.9	71.6	31.2	10.10	51.2	.00251
33	380	500	122.0	86.0	68.3	31.2	9.17	54.3	.00290
34	122	500	122.0	88.0	66.2	31.3	8.64	56.2	.00319
35	18	500	122.0	90.2	63.6	31.2	8.06	58.6	.00356
36	974	100	111.5	63.2	83.8	34.8	12.20	28.0	.00562
37	120	500	121.8	91.2	62.7	31.2	7.79	59.6	.00375
39	2000	500	122.2	82.7	71.8	31.2	10.04	51.0	.00249
40	10,000	500	122.3	81.0	73.5	31.3	10.47	49.3	.00231
41	2200	500	122.5	80.4	74.0	31.2	10.65	48.8	.002245

TABLE VIII DETAILS OF JOINT CONDUCTANCE CALCULATIONS

RUN NO.	CONTACT RESISTANCE (Internal R=.0016)	CONTACT CONDUCTANCE (BTU/hr-ft^2-°F)	CONTACT RESISTANCE (Internal R=.0022)	CONTACT CONDUCTANCE (BTU/hr-ft^2-°F)
3	.00437	228.	.00377	265.
4	.00254	393.	.00194	266.
5	.00260	384.	.00200	500.
6	.00143	699.	.00083	1204.
7	.00260	384.	.00200	500
8	.00687	145.	.00847	118.1
9	.00448	223.	.00608	164.5
10	.00277	361.	.00437	229
11	.00785	127.4	.00725	138.
12	.01827	54.7	.01767	56.6
13	.01437	69.6	.01377	72.6
14	.01085	92.2	.01025	97.6
15	.00771	129.	.00711	140.6
16	.00458	218.	.00398	251.
17	.00308	324.	.00248	403.
18	.00219	456.	.00159	629.
19	.00148	676.	.00088	1136.
20	.00397	252.	.00377	296.
21	.00591	169.	.00531	188.3
22	.01073	93.2	.01013	98.7
23	.00650	154.	.00590	169.5
24	.00244	410.	.00184	544.
25	.00216	464.	.00156	641.
26	.01835	545.	.00124	807.
27	.00168	618.	.00102	980.
28	.00144	696.	.00084	1191.
29	.00133	751	.00073	1370.
30	.00129	771	.00070	1429
31	.000858	1166	.00026	3840
32	.000909	1100	.00031	3230
33	.00130	768	.00070	1428
34	.00158	630	.00099	1010
35	.00196	509	.00136	735
36	.00182	548	.00342	292
37	.00215	465	.00155	645
38	-	-	-	-
39	.000889	1125	.00029	3448
40	.000707	1414 ← 614	.00011	9091
41	.000645	1550	.00005	20000



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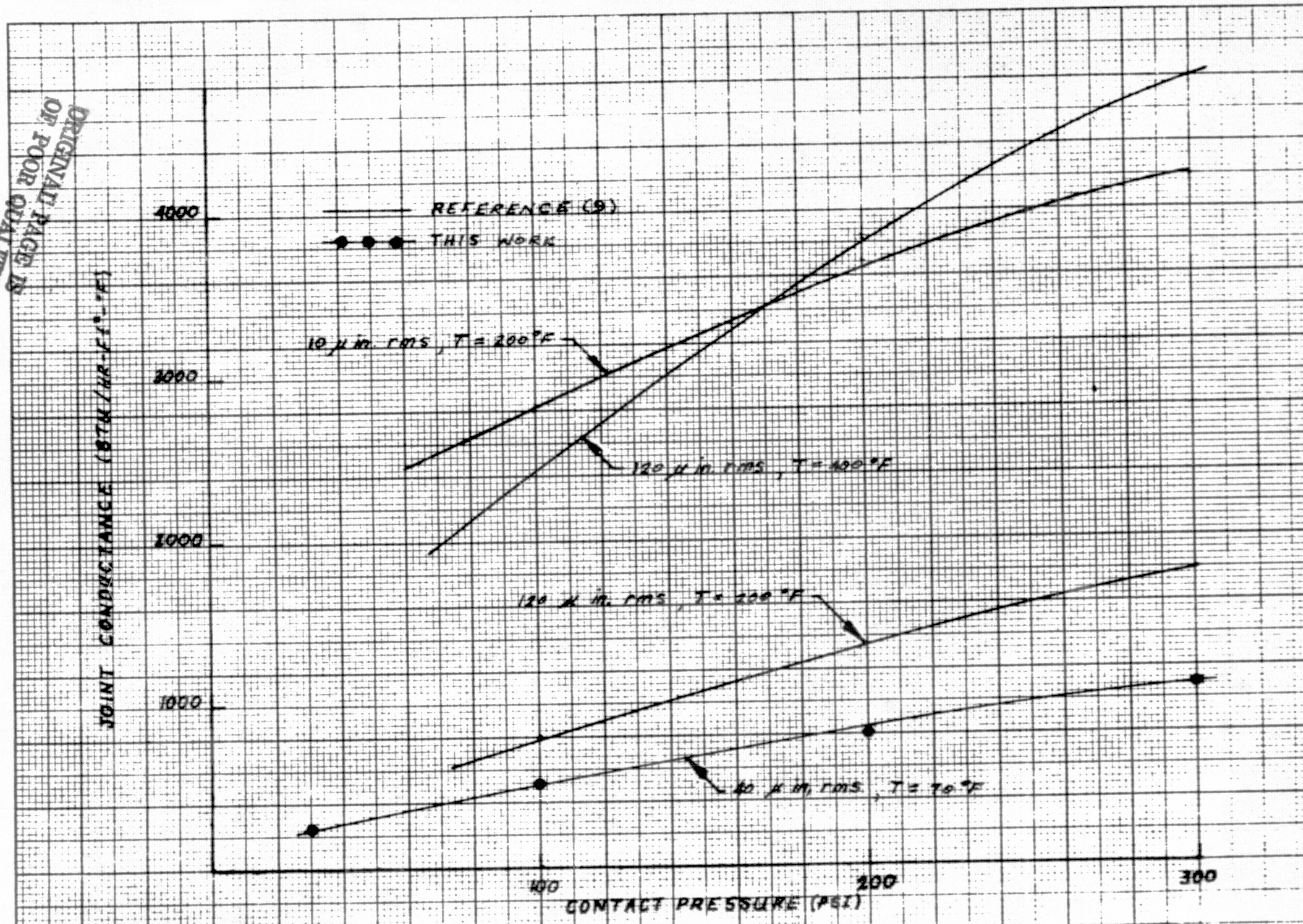

FIGURE 9 EFFECT OF SURFACE ROUGHNESS ON BARE METAL ALUMINUM JOINTS

the expected band for 40μin. rms because large scale waveness and effects of uneven loading reduce the contact area available for heat transfer.

5.0 CONCLUSIONS AND RECOMMENDATIONS

Figure 10 from Reference 1 shows the effect of the contact conductance on the size of the SHRM contact heat exchanger. Since the contact thermal resistance acts in series with resistances internal to the heat exchanger core the total thermal resistance and thus the heat exchanger area are not reduced significantly after the joint conductance exceeds about 1000 BTU/hr-ft^2-°F. Thus values in excess of 1000 BTU/hr-ft^2-°F may be considered to be sufficient for the SHRM application. The test results show that such values are achievable at moderate contact pressures with thermal grease interstitial layers. Thus it is recommended that the SHRM contact heat exchanger be designed assuming that the maximum achievable joint conductance is 1000 BTU/hr-ft^2-°F. This will yield a conservative design of the heat exchanger but will not result in excessive weight penalties. Element tests should be conducted with the final heat exchanger design to insure that the loading mechanism provides contact pressures sufficient to give the necessary joint conductance.

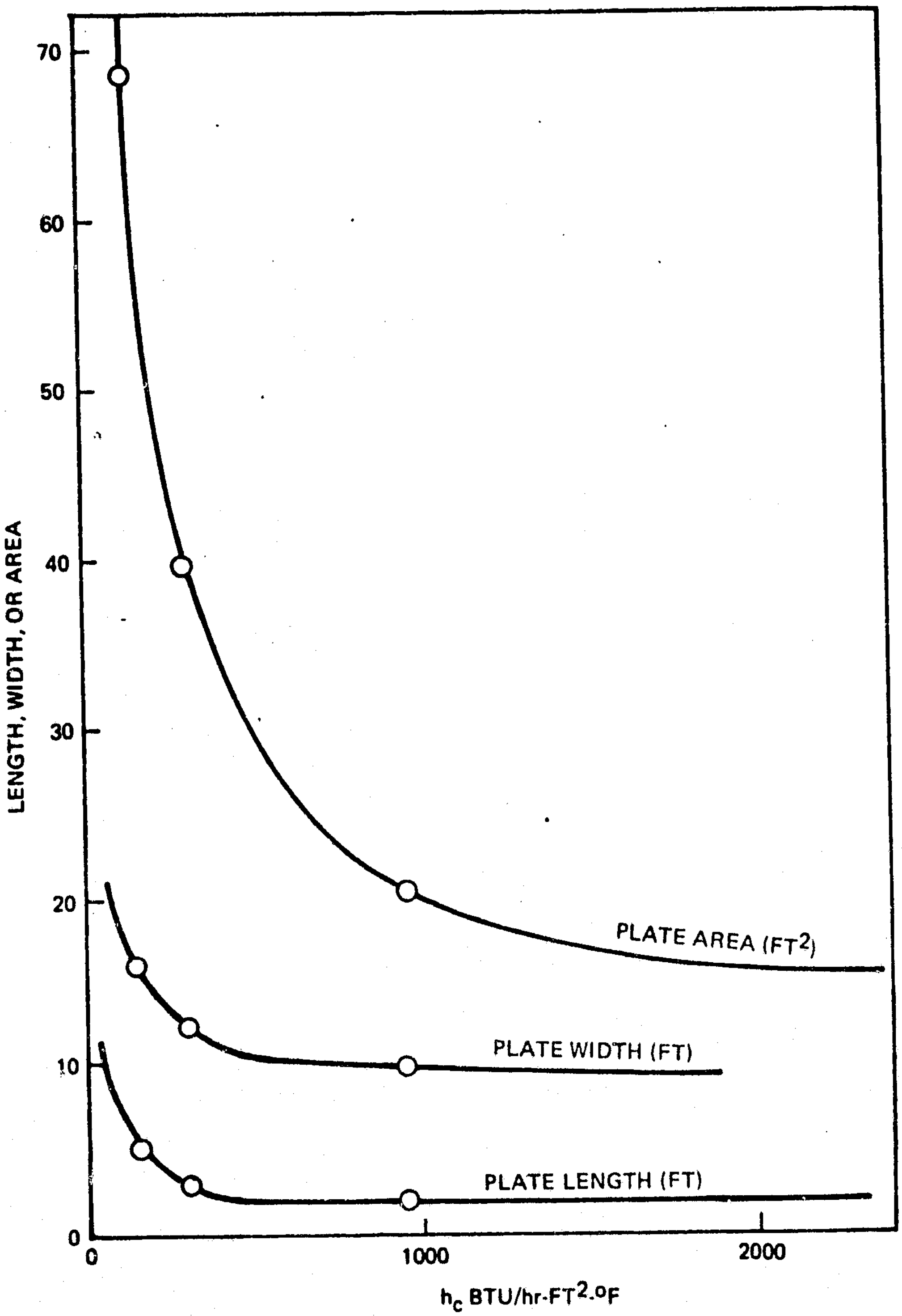

FIGURE 10 EFFECT OF CONTACT CONDUCTANCE ON HX ENVELOPE

6.0 REFERENCES

1) Technical Proposal, "Development of a Self-Contained Heat Rejection Module", VSD report No. 00.1599, May 1973.

2) Venart, J. E. S. and V. K. Magapu, "Thermal Contact Conductance of Irregular Metallic Surfaces",AIAA Paper No. 74-696, July 1974.

3) McKingie, D. J. ,"Experimental Confirmation of Cyclic Thermal Joint Conductance", AIAA Paper No. 70-853, July 1970.

4) Cassidy, J. F., and H. Mark, "Thermal Contact Resistance Measurements at Ambient Pressures of One Atom to 3 x 10^{-12} mmHg and Comparison with Theoretical Predictions", AIAA Paper No. 69-629, June 1969.

5) Cecco, V. S. and Yovanovich, M. M., "Electrical Measurement of Joint Resistance at Perfect Contact Interfaces; Applications to Thermal Conductance", AIAA Paper No. 72-19, January 1972.

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8) Fletcher, L. S., "Thermal Conductance of Gasket Materials for Spacecraft Joints", AIAA Paper No. 73-119, January 1973.

9) Barzelay, M. E., "Range of Interface Thermal Conductances for Aircraft Joints" NASA TN D-426, May 1960.

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APPENDIX I

The coldplates tested in this work contain two sections of hardway flow with a combined length of approximately 2.5" and a section of easyway flow approximately 5.5" long. The internal conductance is obtained by averaging the heat transfer coefficients of these sections according to the following equation

$$\bar{U} = \frac{\Sigma\, \eta_o h A}{A} \qquad \text{(I-1)}$$

The convective heat transfer coefficient in Equation (I-1) may be computed from the generalized performance data in Figure 2

$$h = \frac{j\, \dot{w}\, Cp}{A_c P_r^{2/3}} \qquad \text{(I-2)}$$

The Reynolds number required in Figure 2 is given by

$$Re = \frac{\dot{w}\, D_H}{A_c \mu} \qquad \text{(I-3)}$$

The Freon 21 property data required in Equations (I-2) and (I-3) are taken at an average temperature of 70°F.

μ = 0.780 lb/ft-hr

Cp = 0.250 BTU/hr-°F

$Pr^{2/3}$ = 2.10

For easy way flow at $\dot{w}$ = 500 lb/hr

Re = 2820

j = 0.0115

h = 375

The fin efficiency taken from Figure I-1 is

$$\eta_o = 0.691$$

For hardway flow at $\dot{w} = 500$ lb/hr

$$Re = 2840$$
$$j = 0.022$$
$$h = 1320$$
$$\eta_o = 0.538$$

The average conductance computed from Figure I-1 is

$$\bar{U} = \frac{(2.5)(710) + (5.5)(260)}{8} = 400 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

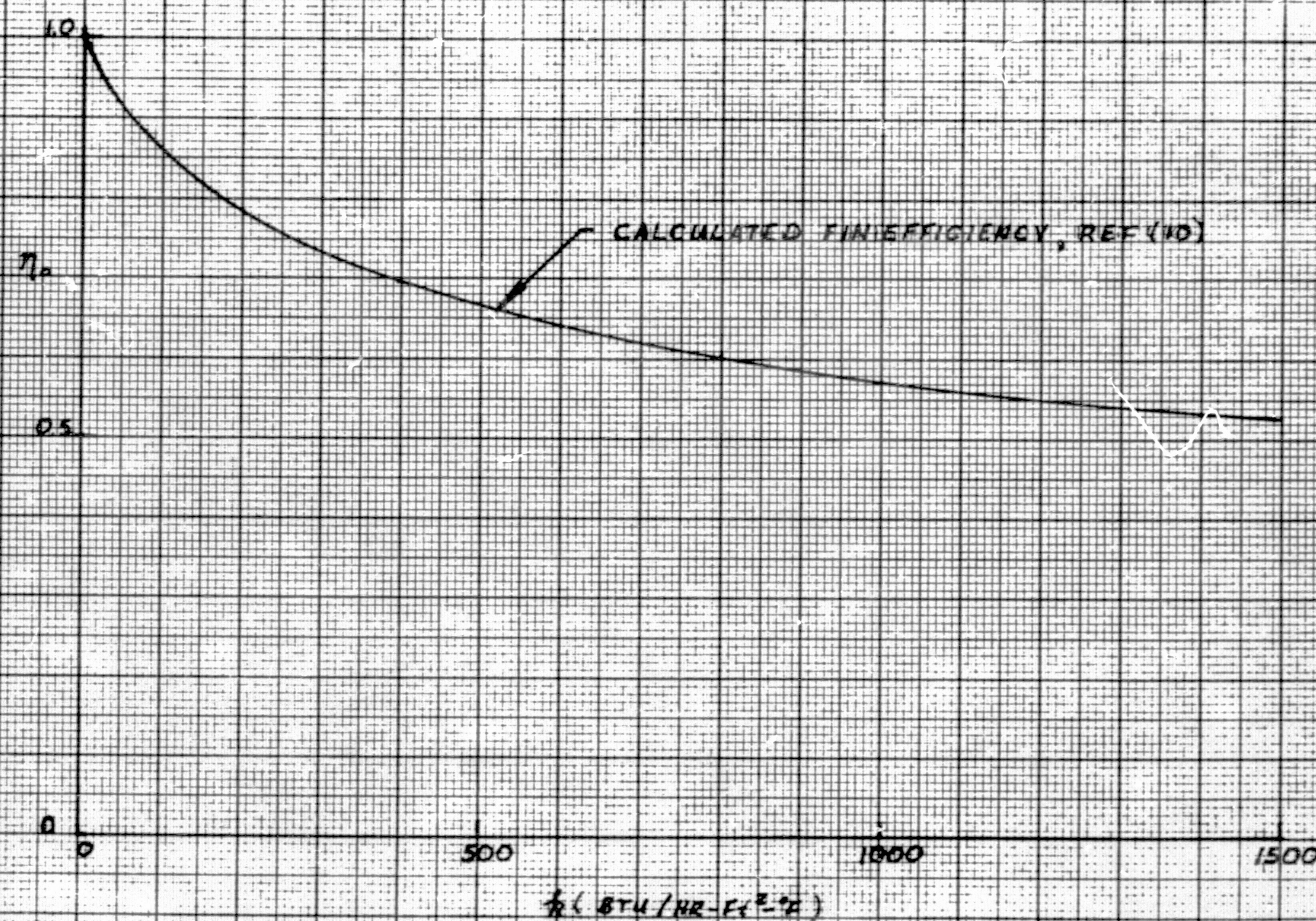

FIGURE I-1 COLDPLATE OVERALL FIN EFFICIENCY

APPENDIX I

DEVELOPMENT OF A SELF-CONTAINED
HEAT REJECTION MODULE

CONTRACT NAS9-13533
REPORT NO. T211-RP-010

PROGRESS REPORT FOR PERIOD
9 July through 27 September 1974

27 September 1974

Submitted By

VOUGHT SYSTEMS DIVISION
LTV Aerospace Corporation
Dallas, Texas

To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
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1.0 INTRODUCTION AND SUMMARY

This eighth progress report under Contract NAS9-13533 (Reference [1]) summarizes Vought Systems Division (VSD) activity during the period of 9 July 1974 through 27 September 1974. The effort during this period has been primarily directed toward re-design and fabrication of a fluid swivel which will not leak at cryogenic temperatures. Another major effort was the testing of two coldplates to establish the joint conductance which can be obtained in a full-scale contact heat exchanger.

2.0 WORK ACCOMPLISHED DURING THIS REPORTING PERIOD

2.1 Fluid Swivel Fitting Testing

Fluid swivel tests were begun on 30 July 1974, having been delayed 3 months while parts were returned to vendor for rework (to stop leakage). Testing was begun in accordance with test request T122-TR-006, as illustrated in Figures 1, 2 and 3. Leakage was checked under dynamic and static conditions for a range of temperatures.

On the first test run, under decreasing temperature, the swivel (S/N 3) began to leak at 227.8°K (-50°F), and continued leaking as temperature was lowered to 172.2°K (-150°F). Leakage rates were 1 g/s (8 lb/hr) dynamic and 0.38 g/s (3 lb/hr) static. On warm-up, static leakage stopped at 227.8°K (-50°F); dynamic leakage continued until a temperature of 283.7°K (50°F) was reached. Very slight leakage was observed during transient temperature increase to 338.9°K (150°F), and none was observed at 338.9°K (150°F). The fitting was rotated throughout the test, except when checking static leakage.

Teardown and inspection of the swivel revealed grease in the seal area and contamination which appeared to be precipitation-type residue left after the grease froze, and was thawed out.

The unit was cleaned and re-assembled using new lip seals, and was re-tested. The results were the same except that the leakage started at 200°K (-100°F), instead of 227.8°K(-50°F). The unit was dimensionally inspected, and found to be per the drawing. Surface finish on the shaft was quite good. The lip seals showed no defects. It was concluded that the unit, as designed, would not seal below 200°K (-100°F).

New lip seals were constructed using thinner stock, in an attempt to increase the compliance of the seal to pressure loading. No improvement in performance could be achieved during subsequent testing.

A search of the specialized seal market identified a line of seals produced by Aeroquip, called Omniseal, that have been used in cryogenic service. After consulting with Aeroquip, several of the seals were obtained. Although not the optimum configuration, it was decided to try them in test. One of the existing swivel fittings was modified to accept the Aeroquip Omniseals. The unit was tested over a temperature range from 150°K (-190°F) to 338.9°K (150°F). Due to a failure in the test set-up, the test was repeated. No leakage was detected at any time on either test run.

On the strength of this test, Omniseals of the precise configuration recommended by Aeroquip were placed on order. A swivel fitting design has been

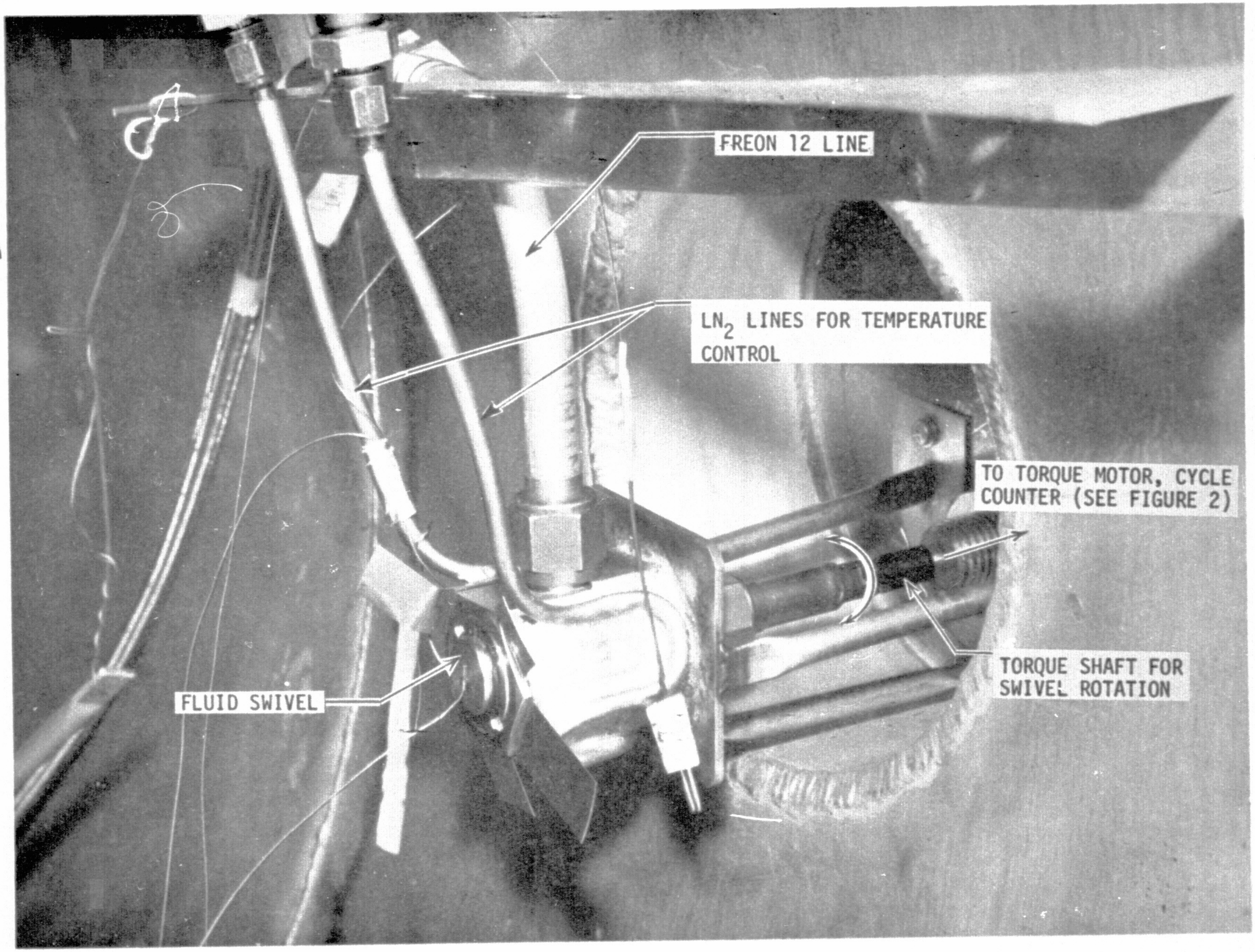

FIGURE 1 IN CHAMBER SWIVEL TEST SETUP

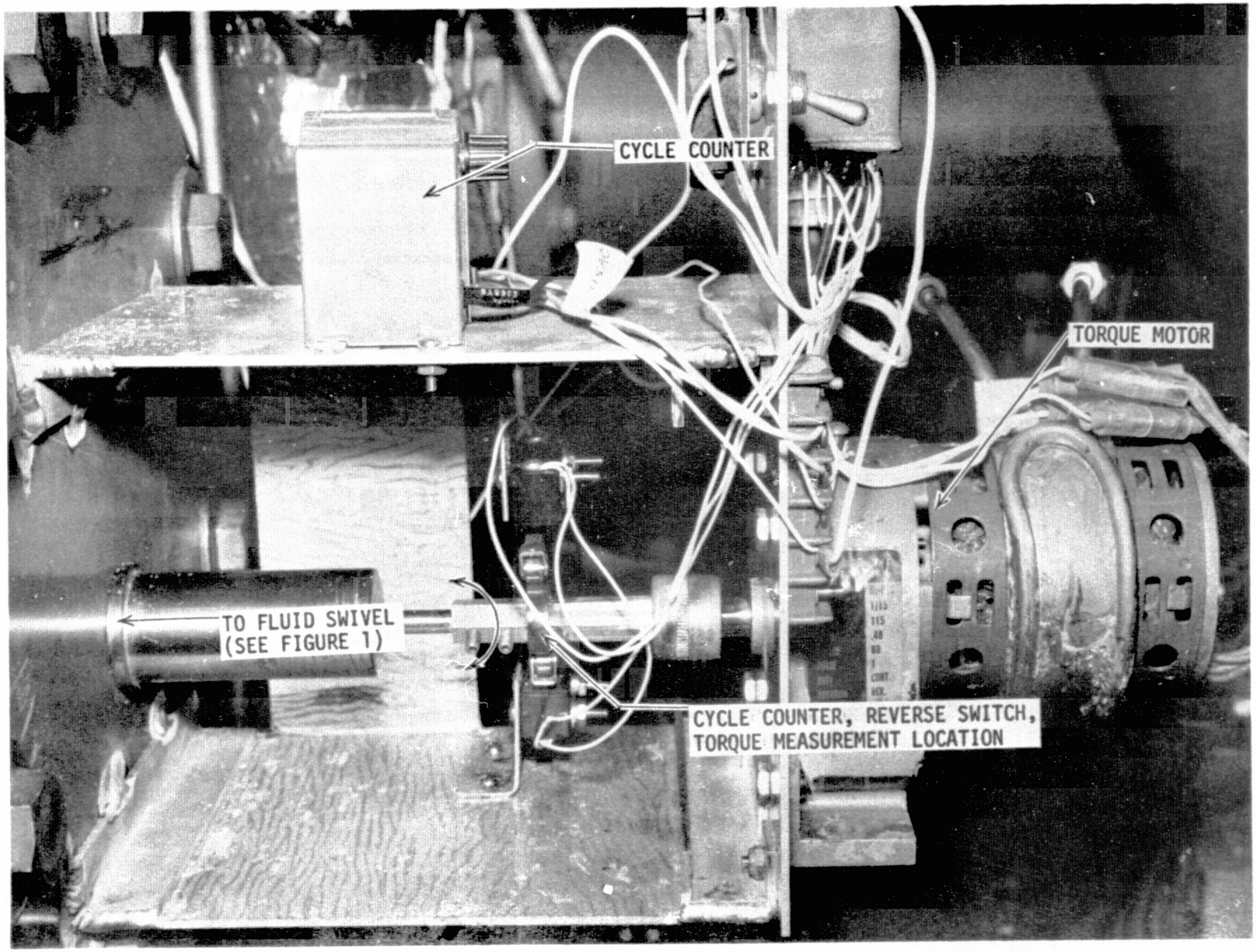

FIGURE 2 OUT CHAMBER SWIVEL TEST SETUP

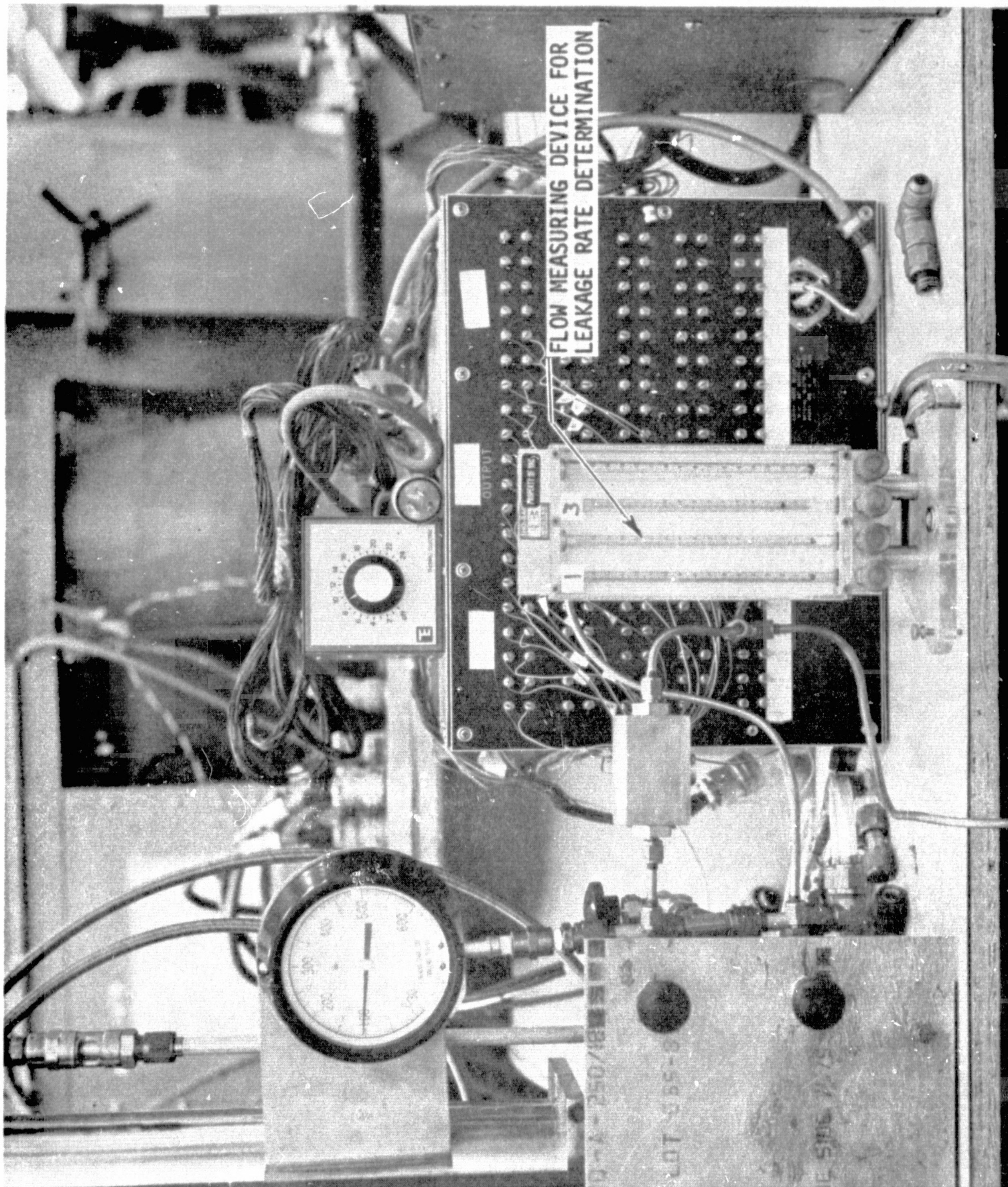

FIGURE 3 FLOW INSTRUMENTATION

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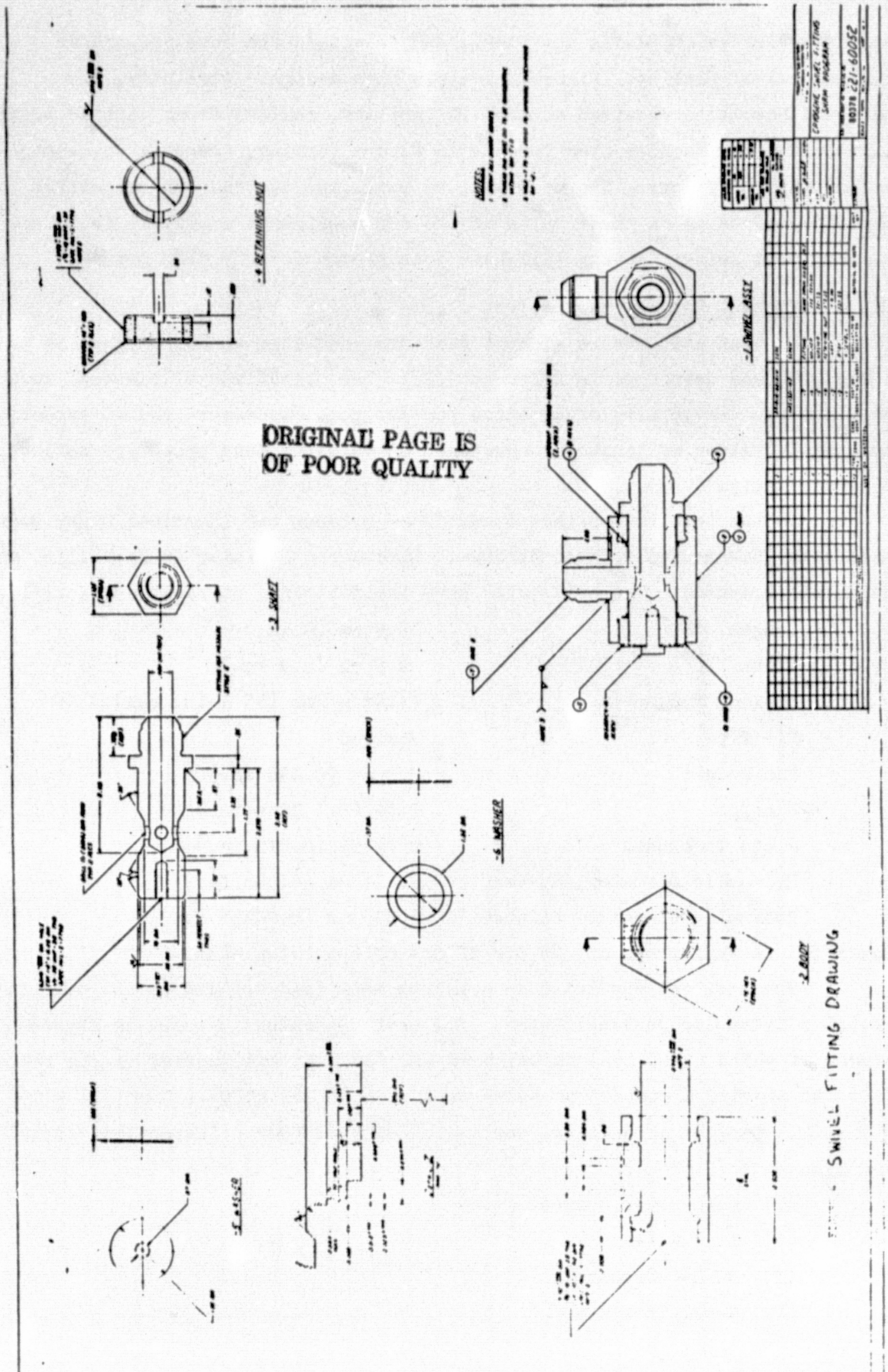

SWIVEL FITTING DRAWING

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rendered which accommodates this seal, and is within the envelope of the original swivel fitting. Figure 4 depicts this design. Currently, the Omniseals have been received at the VSD facility, and the first unit is being built in the VSD Machine Shop for use in future testing. Pending successful test of this first unit, two more will be built and tested. On successful completion of tests on three units of the new configuration, it is felt that an acceptable swivel fitting will have been proven for the SHRM program.

2.2 Contact Heat Exchanger Test

A joint conductance element test was conducted at VSD between 24 to 31 May 1974, as described in Reference [2]. The objectives of the test were to establish the feasibility of a bolted contact heat exchanger, and to establish appropriate values of contact conductance on which to base prototype contact heat exchanger design.

In the test the contact conductance between two identical Lunar Module coldplates (fabricated by Avco Aerospace Structures Division in Nashville, Tennesse) was measured. The coldplates have the following physical characteristics:

Length	22.9 cm (9 in.)
Width	9.9 cm (3.9 in.)
Surface Roughness	1.143 μ rms (45 μ-in. rms)
Fin Type	Pierced
Fin Height	2.3 mm (0.090 in.)
Fins/cm	2.64 (6.7 fins/in.)
Plate Thickness	0.76 mm (0.030 in.)
Hydraulic Diameter, Easyway	2.438 mm (0.008 ft.)
Hydraulic Diameter, Hardway	1.335 mm (0.00438 ft.)

Figure 5 is a radiograph of the top of one of the coldplates.

The test was conducted in a vacuum with load applied to the coldplates through a hydraulic jack mechanism. The test apparatus is shown in Figures 6 and 7. A force of 445 N (100 lb_f) to 44.48 kN (10,000 lb_f) was applied to the two coldplates to provide a contact pressure variation of 207 kN/m^2 (30 psi) - 2.068 MN/m^2 (300 psi). Contact conductance was evaluated with four different interstitial conditions.

(1) Bare Metal Contact

(2) Lead Foil

(3) DC-340 Grease

(4) G-641 Grease

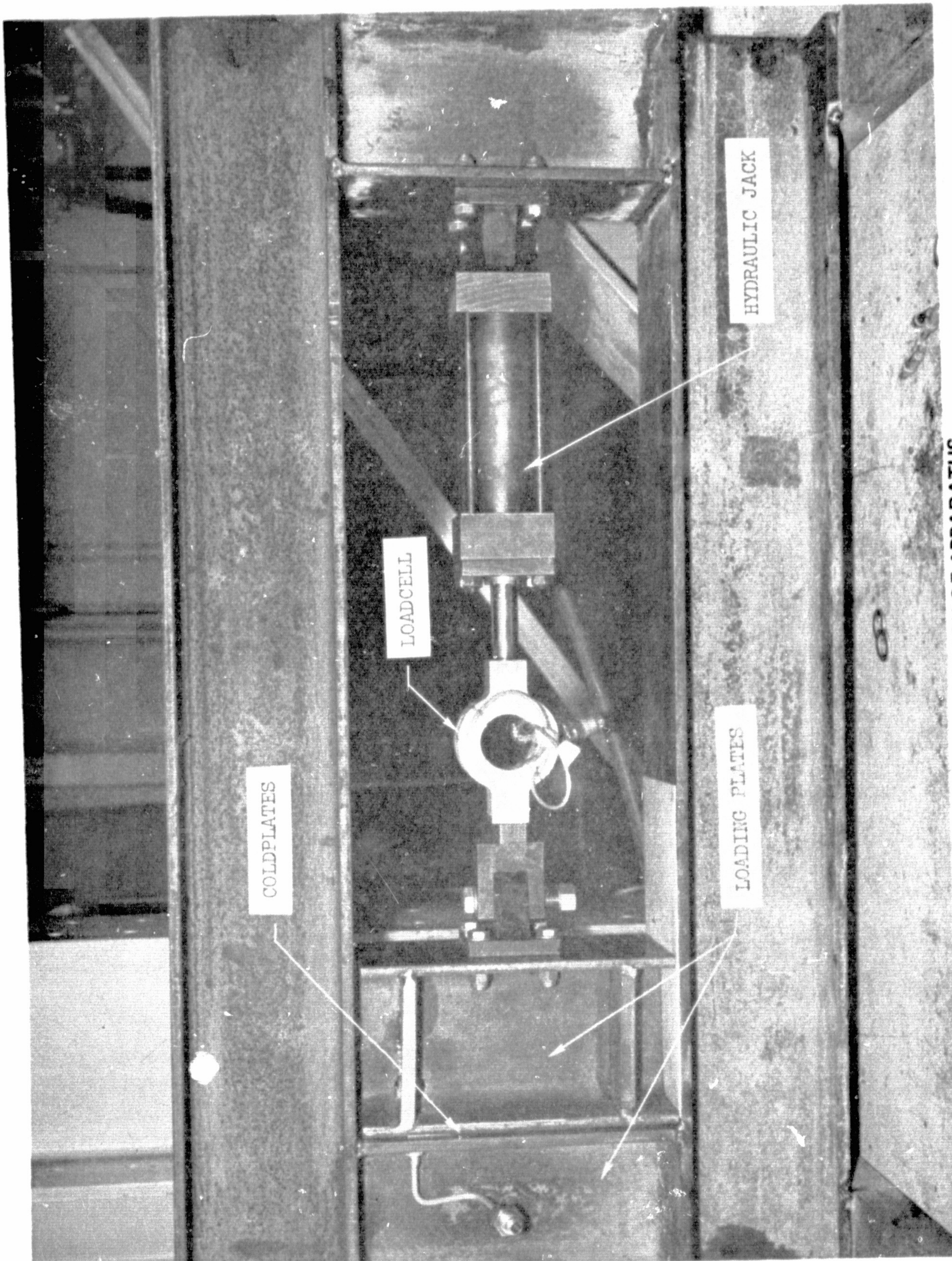

FIGURE 6 PHOTOGRAPH OF APPARATUS

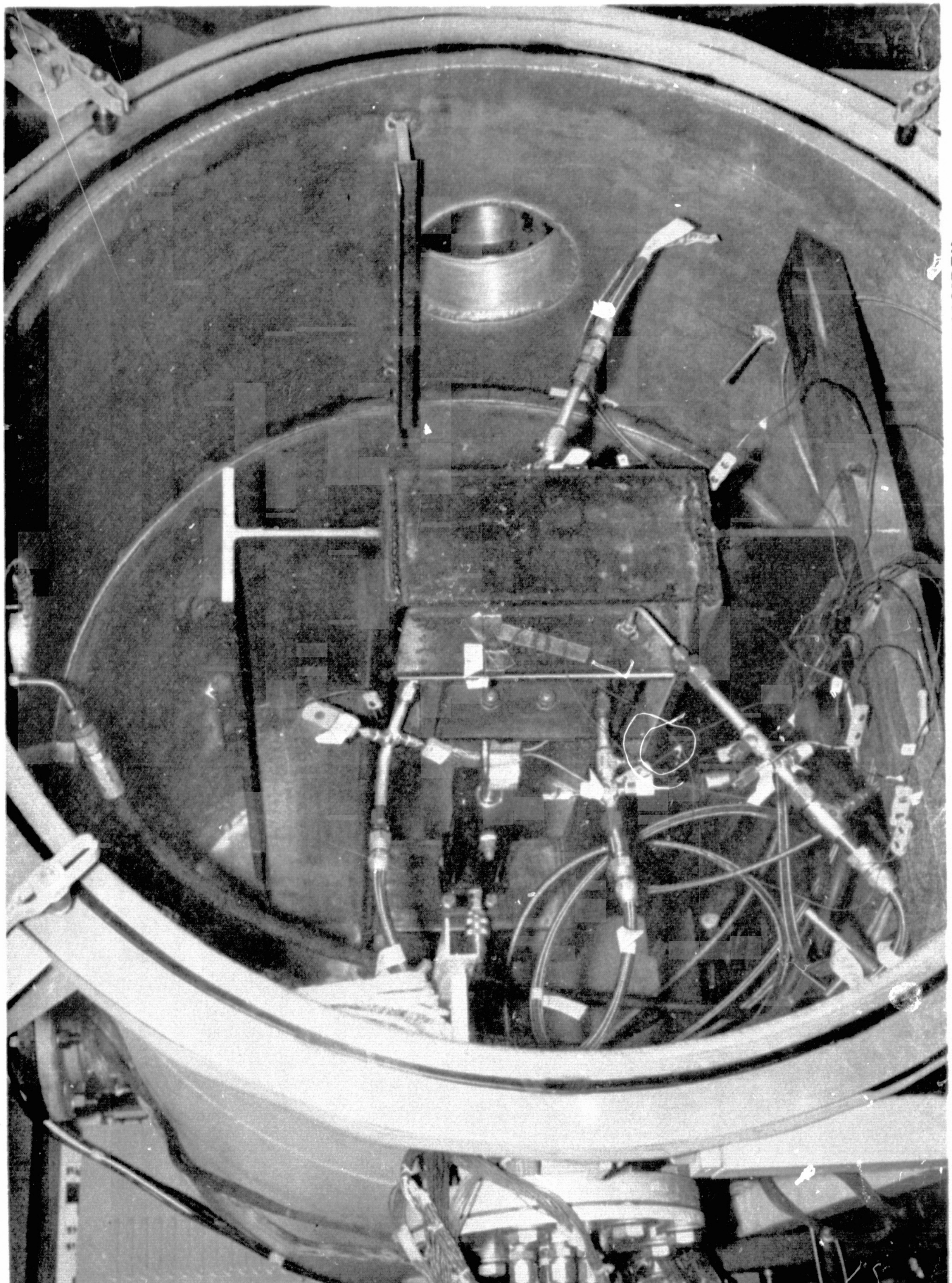

FIGURE 7 INSTALLATION OF TEST ARTICLE IN VACUUM CHAMBER

Results were in agreement with, but were lower than previously published vacuum contact conductance data for coupon specimens. This was expected because of the large-scale waviness of the relatively large area of the coldplates. Figure 8 shows the contact conductance values obtained compared to previously published data (Reference [3]). These results are significant to contact heat exchanger design in that they show the level of contact conductance that can be expected, and they show that the use of a thermal grease is desirable. A value of contact conductance in the range of 3 $kJ/m^2 s^\circ K$ (500 $BTU/hr\text{-}ft^2\text{-}^\circ F$), which is desirable, can most reliably be achieved over a wide range of pressures (which will permit large pressure level tolerances) with a thermal grease interstitial filler material.

2.3 Contact Heat Exchanger Design

Design of contact heat exchangers (HX) has been discussed previously in References [4] - [6]. A contact HX contact conductance of 2.3 $kJ/m^2 s^\circ K$ (400 $BTU/hr\text{-}ft^2\text{-}^\circ F$) was established for initial sizing exercises. Based on the contact HX element tests it appears that a joint contact conductance of 4.6 $kJ/m^2 s^\circ K$ (800 $BTU/hr\text{-}ft^2 {}^\circ F$) seems reasonable if use of a thermal grease is feasible.

Originally, a contact HX design was evolved using 20 coldplates, 10 on each side of the HX, as shown in Figure 4. It was based on U_c = 2.3 $kJ/m^2 s^\circ K$ and a core as shown in Table I. Characteristics of the HX are shown in Table II. This design appears reasonable; however, it is expensive due to the number of individual pieces (coldplates) and difficulty in manifolding 10 coldplates for each side of the HX. One advantage of fewer plates is that it is relatively easier to control the waviness of the plates.

To reduce costs and to incorporate the results of the contact HX element tests, the design conditions for the contact HX were modified, as shown in Table III. In the revised design 6 coldplates are used rather than 20, and individual connectors are used on each coldplate rather than using a common header. To accommodate this change the allowable pressure drop was increased from 27.6 kN/m^2 (4 psi) to 55.2 kN/m^2 (8 psi) on the freon side, and from 13.8 kN/m^2 (2 psi) to 27.6 kN/m^2 (4 psi) on the water side. This increase in ΔP reduces the amount of HX area required, but has no significant system impact since the liquid-phase pump was sized for a larger pressure head than is planned for the system. In the refrigeration mode some degradation in system performance may be incurred, however, it is small (about 1% efficiency) and it is predictable.

The HX design was established using VSD's HEDR computer routine

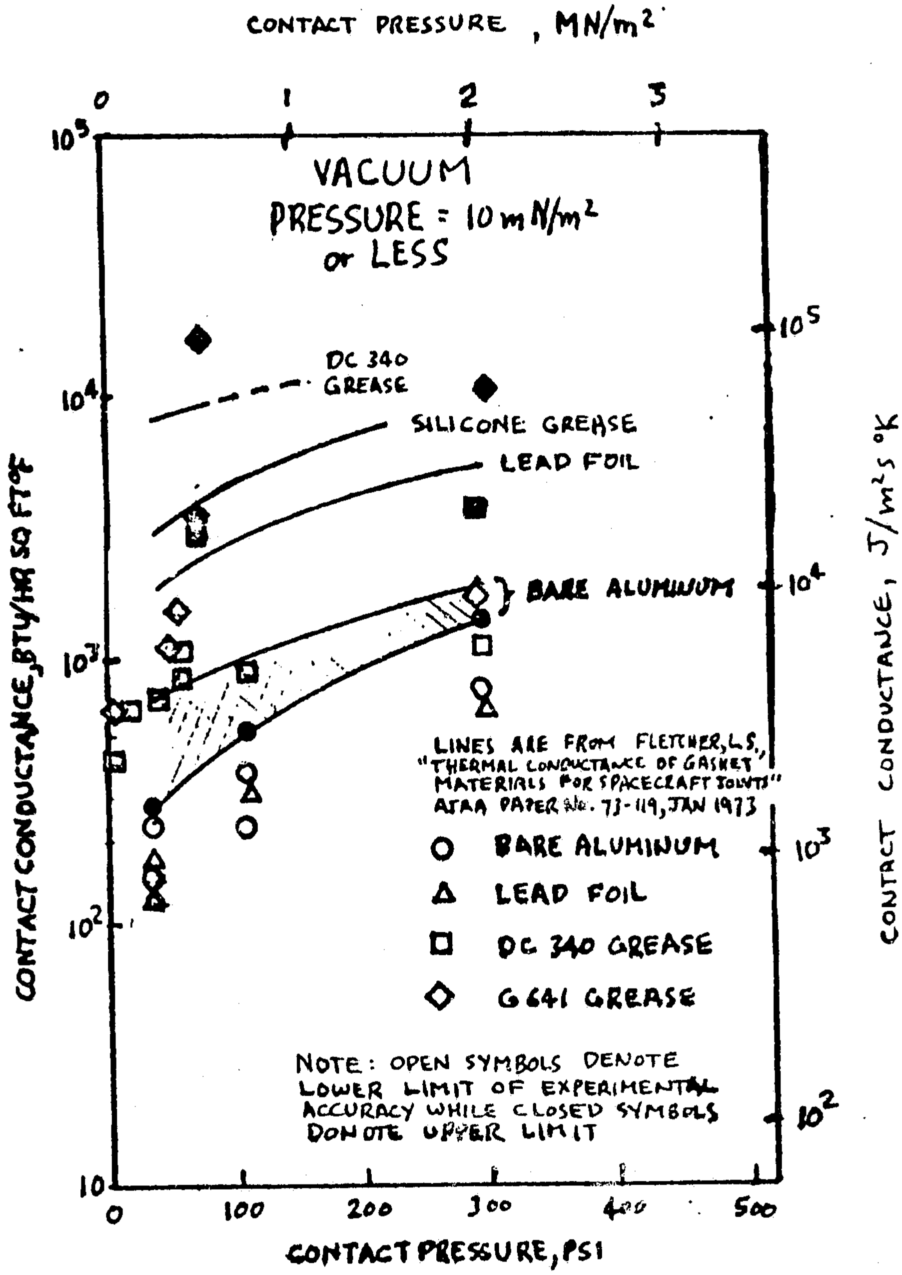

FIGURE 8

COMPARISON OF CONTACT CONDUCTANCE FROM
TEST WITH PREVIOUS DATA

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TABLE I

CONTACT HX CORE CHARACTERISTICS

Fin Type	Lance
Fin Height	2.54 mm (0.1-in.)
Fin Spacing	7.3 fin/cm (18.5 fins/in.)
Fin Thickness	0.127 mm (0.005-in.)
Offset Length	3.18 mm (1/8-in.)
Plate Thickness	1 mm (0.040-in.)
Plate Waviness	1 μ rms (40 μ-in. rms)

270.

TABLE II

ORIGINAL CONTACT HX DESIGN CONDITIONS

Heat Transfer Rate	=	14.65 kW (50,000 BTU/hr)
Effectiveness, ε	=	0.867
Contact Pressure	=	≥ 2.07 MN/m^2 (300 psi)
Filler Material	=	None

	HOT SIDE (WATER)	COLD SIDE (FREON 12)
Flowrate, g/s(lb/hr)	98.8 (785)	403 (3200)
Inlet Temp., °K (°F)	313.9 (105)	272.2 (30)
Outlet Temp., °K (°F)	227.8 (40)	308.3 (95)
Operating Pressure MN/m^2(psi)	0.345 (50)	2.07 (300)
Pressure Drop kN/m^2 (psi)	3.45 (0.5)	27.6 (4)

TABLE III

REVISED DESIGN CONDITIONS

Heat Transfer Rate	=	14.65 kW (50,000 BTU/hr)
Effectiveness, ε	=	0.867
Contact Pressure	=	345 kN/m^2 (50 psi)
Filler Material	=	G641 Thermal Grease

	HOT SIDE (WATER)	COLD SIDE (FREON 12)
Flowrate, g/s (lb/hr)	98.8 (785)	403 (3200)
Inlet Temp., °K (°F)	313.9 (105)	272.2 (30)
Outlet Temp., °K (°F)	227.8 (40)	308.3 (95)
Operating Pressure MN/m^2(psi)	0.345 (50)	2.07 (300)
Pressure Drop kN/m^2 (psi)	13.8 (2)	55.1 (8)

NOTE: Fittings are 2.54 cm (1-in.) diameter AN type - MS33656-16

(Reference [7]). The design is shown in Figure 9.

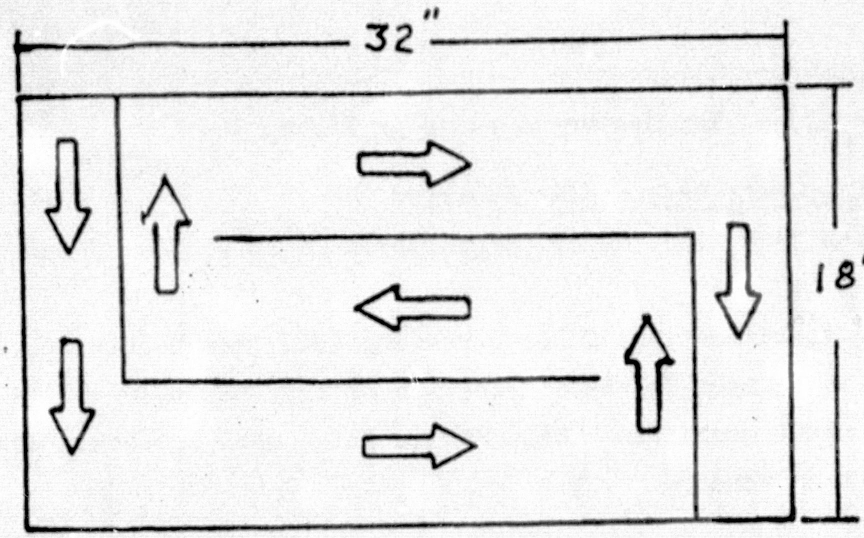
2.4 Component Procurement Activity

The TASK, Inc. motor compressor unit is scheduled for delivery in mid-October 1974.

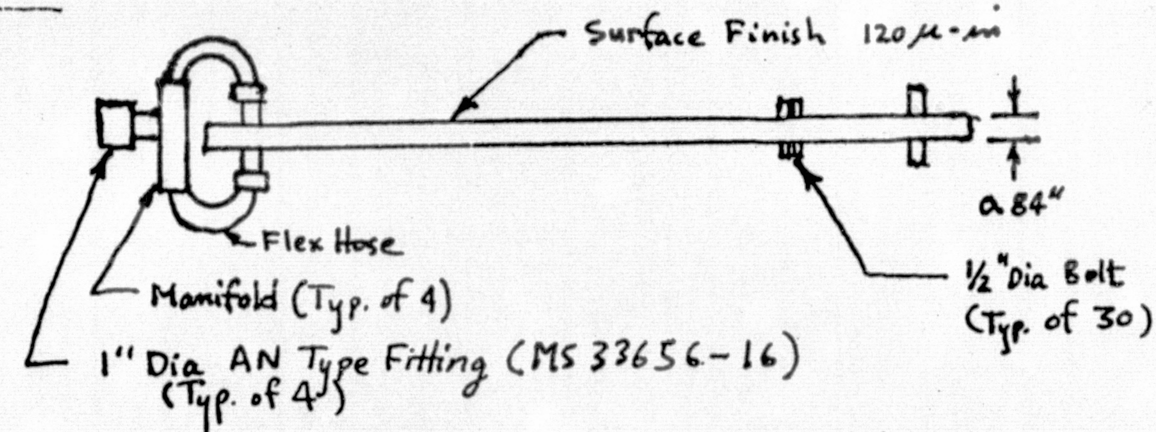
A Specification Control Drawing (SCD) has been prepared for the tube extrusion to be used in the fabrication of four radiator panels for use in the SHRM deployment mechanism. This SCD is being used to obtain quotes and delivery schedules from vendors.

2.5 Laboratory Prototype SHRM Test

Work was initiated on the Laboratory prototype SHRM test plan; however, it was suspended prior to complete definition due to slippage in component delivery schedules.



CONTACT HX CORE LAYOUT
ARROW DENOTES EASYWAY



SIDE VIEW OF CONTACT HX
CONTAINING 6 COLDPLATES

NOTES:

(1) PROOF PRESSURE = 450 psi (Oper. Press. = 300 psi)
(2) WATER HEAT TRANS COEF. = 500 BTU/HR FT^2 °F
(3) FREON 12 HEAT TRANS COEF. = 700 BTU/HR FT^2 °F
(4) CONTACT COEFFICIENT = 800 BTU/HR FT^2 °F
(5) EFFECTIVENESS = 0.87 ; NTU = 8
(6) WEIGHT = 20 LB
(7) MAT'L; ALUMINUM

FIGURE 9

NEW CONTACT HX DESIGN

3.0 CURRENT PROBLEM AREAS

There is currently a technical problem area related to speed control of the motor-compressor. It was planned to use an inverter to vary frequency and voltage, in a linear relationship, to control the motor-compressor speed, and thus the load carrying capacity of the refrigeration system. A common inverter was to be used to drive each of the two candidate motor-compressor units. There is now concern that it may not be feasible to drive the two units with a common inverter. This problem is currently being investigated.

Late component deliveries are still having an impact on the program schedule, and appear likely to cause more extensions in the prototype SHRM test date.

4.0 WORK PLANNED FOR NEXT REPORTING PERIOD

During the next reporting period the following work will be completed.

(1) fabrication and testing of three cryogenic-temperature fluid swivels

(2) preparation of the SHRM Prototype Test Plan

(3) SCD for the contact HX

5.0 REFERENCES

[1] NASA-JSC Contract NAS9-13533, dated 18 June 1973.

[2] Leach, J. W., "Joint Conductance Element Tests for the Self-Contained Heat Rejection Module Contact Heat Exchanger", VSD Report No. T211-RP-009, dated 15 August 1974.

[3] Fletcher, L. S., "Thermal Conductance of Gasket Materials for Spacecraft Joints", AIAA Paper No. 73-119, January 1973.

[4] VSD Report No. 00.1599, "Technical Proposal - Development of A Self-Contained Heat Rejection Module" dated 2 May 1973.

[5] VSD Report No. T211-RP-003, "Self-Contained Heat Rejection Module Progress Report", dated 2 November 1973.

[6] VSD Report No. T211-RP-004, "Self Contained Heat Rejection Module Progress Report", dated 7 December 1973.

[7] Leach, J. W. and Phillips, M. A., "Heat Exchanger Design Routine (HEDR) User's Manual", VSD Report No. 00.1606, 28 May 1973.

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APPENDIX J

PHASE II AND III - DEVELOPMENT OF A SELF-CONTAINED HEAT REJECTION MODULE (SHRM)

Contract No. NAS9-14408
Report No. T211-RP-012

PROGRESS REPORT FOR THE PERIOD
13 December 1974 through 13 January 1975

14 January 1975

Submitted by:

VOUGHT SYSTEMS DIVISION
LTV Aerospace Corporation
Dallas, Texas

To

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Johnson Space Center
Houston, Texas

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ECS Group

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1.0 INTRODUCTION AND SUMMARY

This is the first progress report submitted under Contract NAS9-14408 (Reference [1]); it covers VSD activity for the period of 13 December 1974 through 13 January 1975. The major accomplishments of this period were:

(1) Completion and release of the installation drawing for the SHRM radiator panels.

(2) Release of the contact heat exchanger RFQ to vendors.

(3) Initiation of fabrication of the fluid swivel for use in the radiator deployment mechanism.

(4) Updating of the program schedule.

The most significant problem area which requires NASA-JSC direction is definition of the inverter which drives the refrigeration system compressor. It is necessary to make a decision on whether a solid state inverter or a rotary inverter will be used in the flight prototype test to enable design of the control system to begin on schedule (1 February 1975).

Work planned for the next month includes:

(1) Selection of a contact heat exchanger vendor.

(2) Completion of the fabrication and leak-testing of the fluid swivels.

(3) Completion of the SHRM radiator panels and delivery to NASA-JSC.

2.0 WORK ACCOMPLISHED DURING THIS REPORTING PERIOD

2.1 Program Plan

The program schedule has been prepared to interface with other programs which involve testing in the Space Environment Simulation Laboratory (SESL) at NASA-JSC. The overall schedule is dominated by vacuum test chamber availability at VSD and at NASA-JSC. The program also has to be tailored to support the SHRM simulated zero-g radiator deployment tests at NASA-MSFC. This necessitates design, fabrication, and delivery of the radiator panels early in the program.

Delays in the laboratory prototype test due to hardware delivery time extensions have reduced the opportunity to use data and experience from that test in the flight prototype SHRM design. This increases the technical risk in design, and makes achievement of optimum performance in the flight prototype SHRM less likely. The schedules for the Phase II and III SHRM efforts are given in Figure 1.

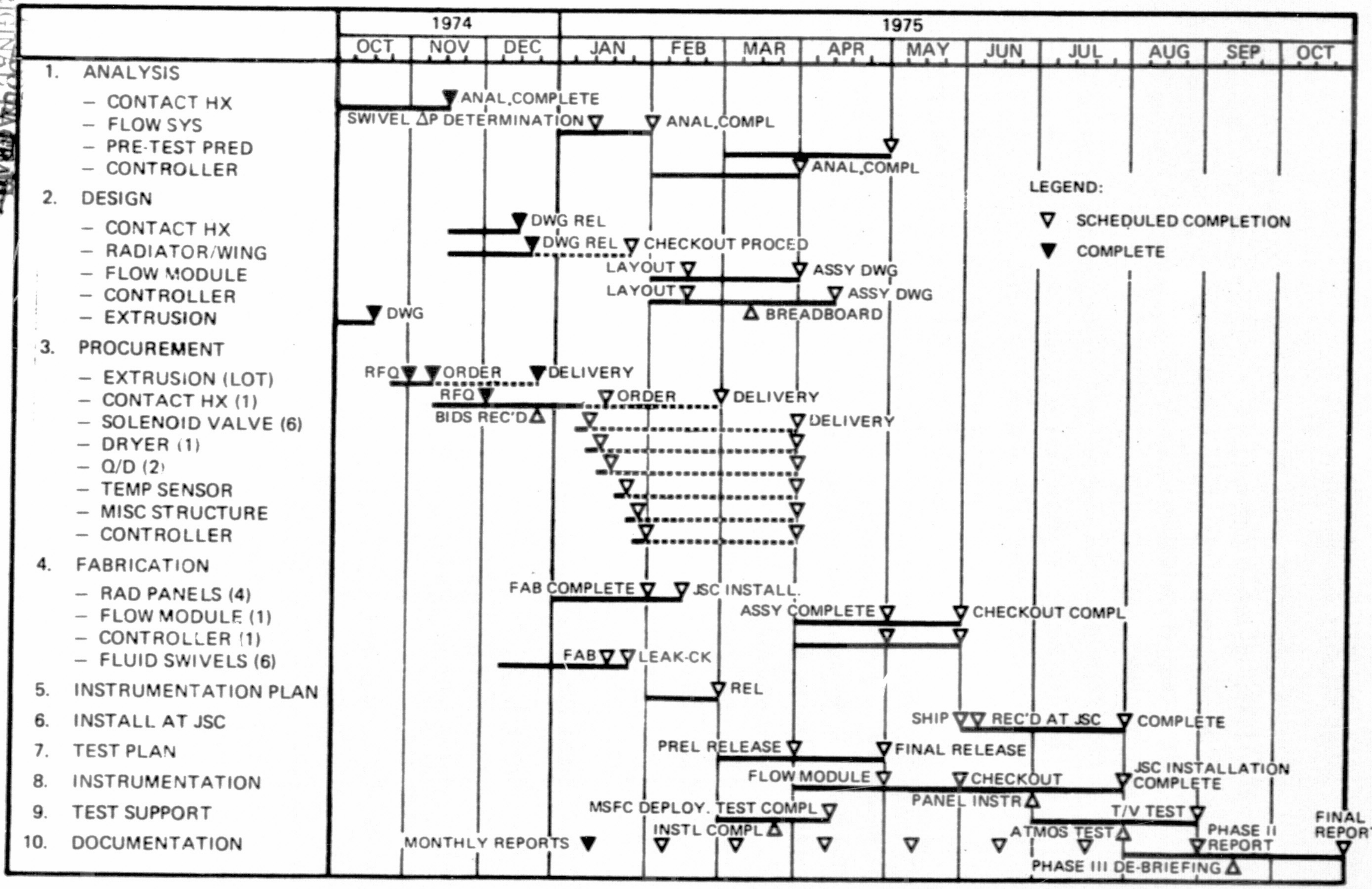

FIGURE 1 SHRM PHASE II AND III SCHEDULE

2.2 Radiator Panel Design

The SHRM radiator panel installation drawing (VSD drawing 221-60056) was released on 2 January 1975. The design was based on the analysis reported in Reference [2]. The general characteristics of the design are shown on Figure 2. One change has been required in the design to facilitate fabrication: the manifold wall thickness has been increased from 0.035 in. to 0.049 in.

2.3 Radiator Panel Fabrication

Fabrication of the four radiator panels for the flight prototype SHRM was initiated on 2 January 1975. All hardware and components for the radiator panels were ordered prior to release of the panel installation drawing. The fabrication effort is currently behind schedule because of late arrival of the aluminum tube extrusions. Attempts will be made to recover from this and to deliver the radiator panels on schedule. It may be necessary to delay calibration of the orifices which balance the flow between panels from the time of manufacture until after return of the panels from MSFC in order to meet the delivery date. This would involve some risk of greater expense when the calibration is actually accomplished.

2.4 Flight Prototype Flow Schematic

Detailed analysis of the flow module for the flight prototype SHRM has been initiated. The flow schematic for the system is shown in Figure 3. The details of the control system operation are being definitized and the solenoid valves used in the mode transfer system are being sized.

2.5 Control System/Inverter Interface

During the refrigeration mode of operation heat load control in SHRM is achieved by adjusting compressor speed. This is done by varying the input voltage and frequency to the a.c. motor driving the compressor. The method of providing the variable voltage and frequency power to the compressor is thus of prime importance to the SHRM control system. The flight system will use a solid-state inverter to convert spacecraft d.v. power to a.c. at the required voltage and frequency. This will require only a signal from the control system which is proportional to the desired compressor speed.

If a solid state inverter is not available for use in the flight prototype test then the rotary inverter being used in the laboratory prototype test will have to be used. The rotary inverter has a manual adjustment on output voltage and frequency, and so cannot be integrated in the control system as the solid

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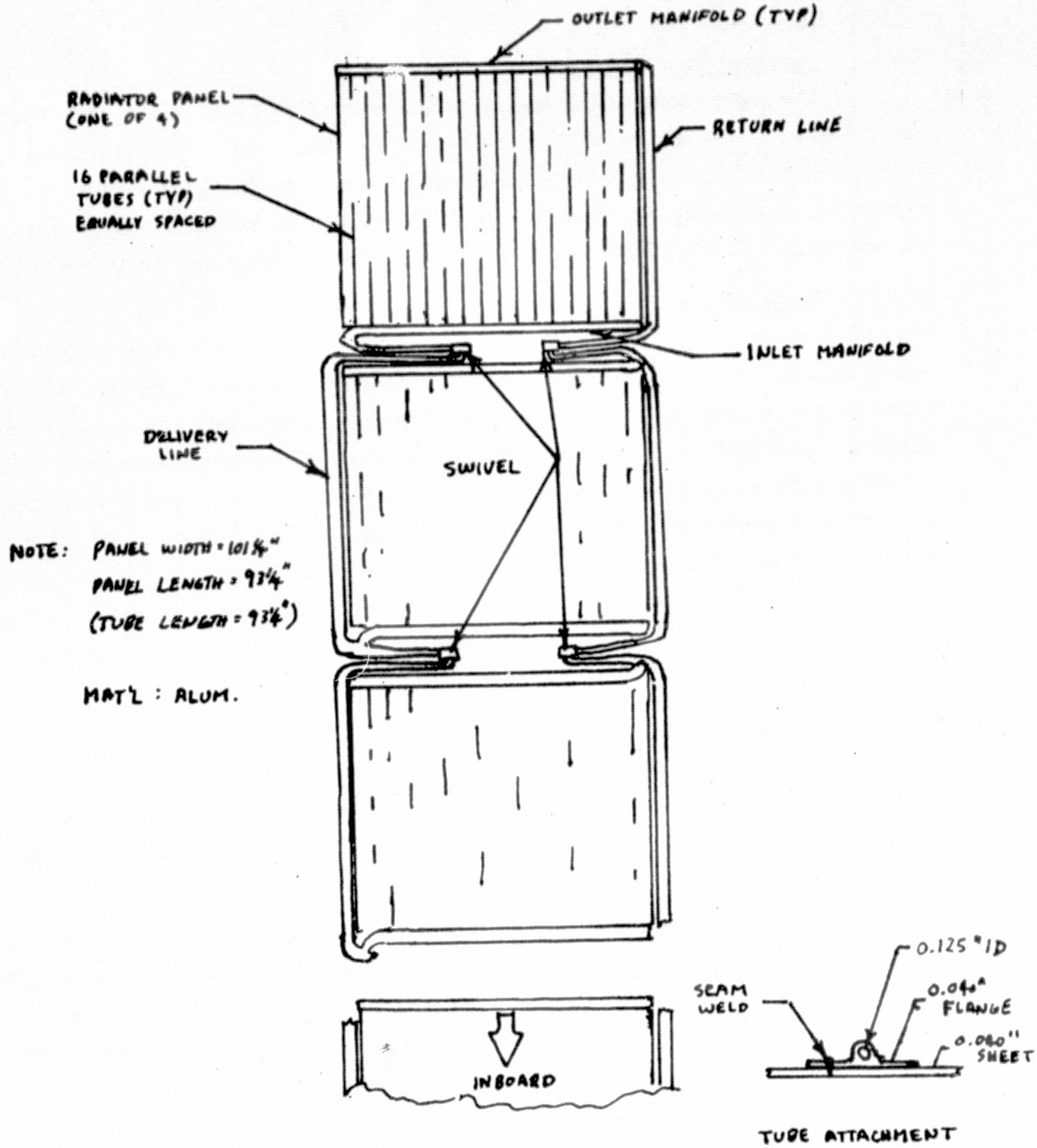

FIGURE 2 SHRM RADIATOR CONFIGURATION

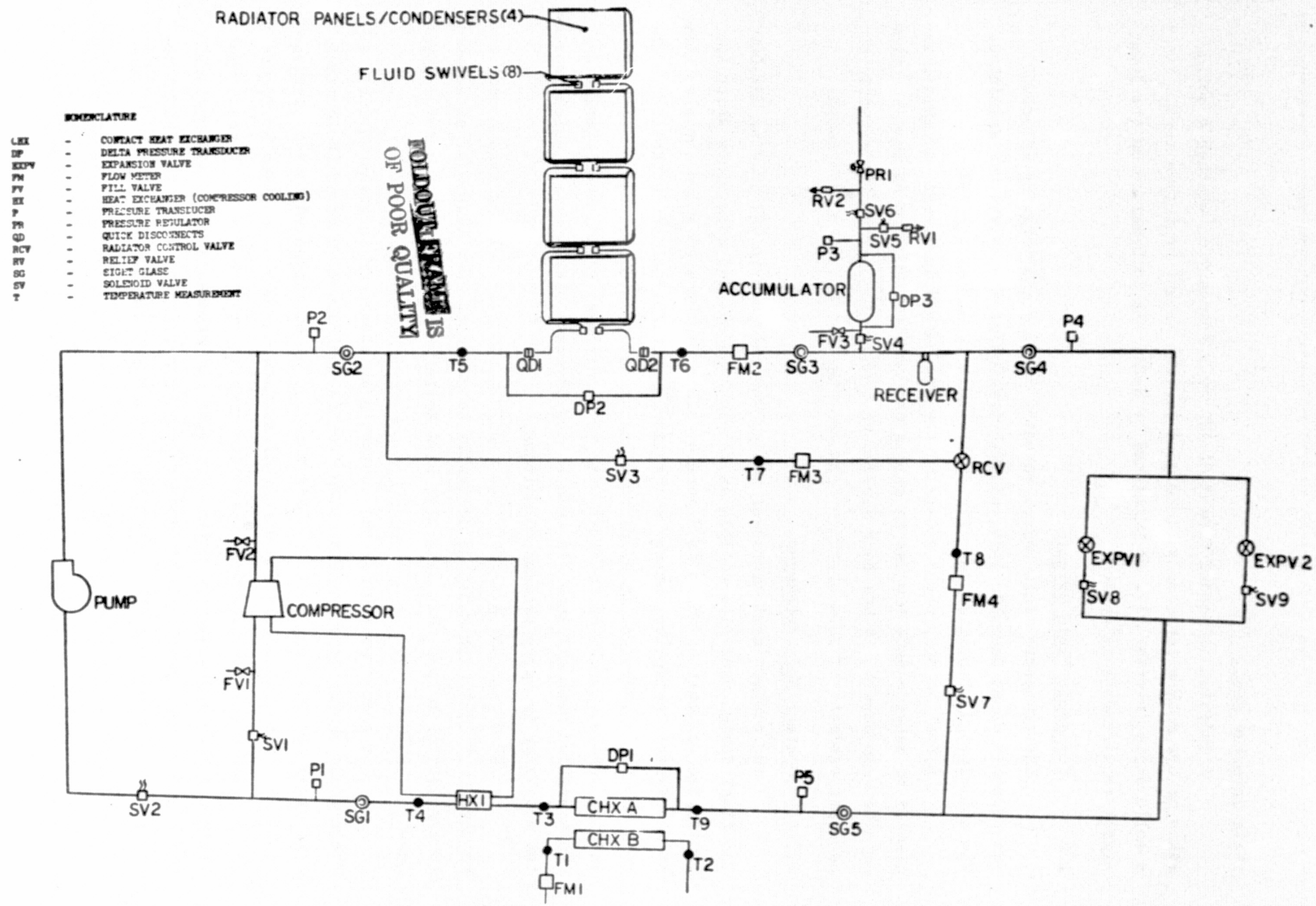

FIGURE 3 SHRM FLOW MODULE SCHEMATIC 1/10/75

state inverter can.

A decision is needed from NASA-JSC by 1 February on whether a solid state or the rotary inverter will be used for the flight prototype test. If a solid state inverter will be furnished by NASA, the information on the input signal requirements is needed.

2.6 Cryogenic Fluid Swivel

The cryogenic fluid swivel assembly drawing (VSD drawing 221-60052) was released on 16 December 1974, and fabrication of the swivel was initiated. The anticipated completion date for the swivels is 31 January 1975.

A test to accurately establish the pressure drop across the swivel was planned to provide data for use in sizing the orifices used to balance the flow-rate between panels. It has been decided that analysis can provide a sufficiently accurate estimate of this pressure drop, so the test has been eliminated.

2.7 Contact Heat Exchanger

The contact heat exchanger design was released in VSD drawing 221-60053. The final design uses five coldplates on each the hot and cold sides which are manifolded together as shown in Figure 4. This figure also shows the plan view of the typical coldplate, which is identical for all 10 of the coldplates. The heat exchanger sides are mated by rotating one coldplate subassembly 180° and sliding the subassemblies together with the individual coldplates interlaced. The assembled heat exchanger is then bolted together with 45 bolts which provide the necessary pressure between coldplates.

The design has a flight-type manifold; and the bolting arrangement could be used in flight. It would be desirable to use a quicker method of attachment in a flight prototype, but this will have to be proven out in later development work.

The specification for the contact heat exchanger is given in Table I. These requirements were largely drawn from the coldplates used in the contact heat exchanger feasibility test (Reference [3]). Additional data correlation from that test considering the effect of surface roughness is given in Figure 5.

An RFQ was released to three prospective vendors for the contact heat exchanger with a due date of 6 January 1975. An extension to 13 January 1975 was requested and granted. Two quotes have been received, with one vendor declining the bid because the coldplates are larger than his brazing

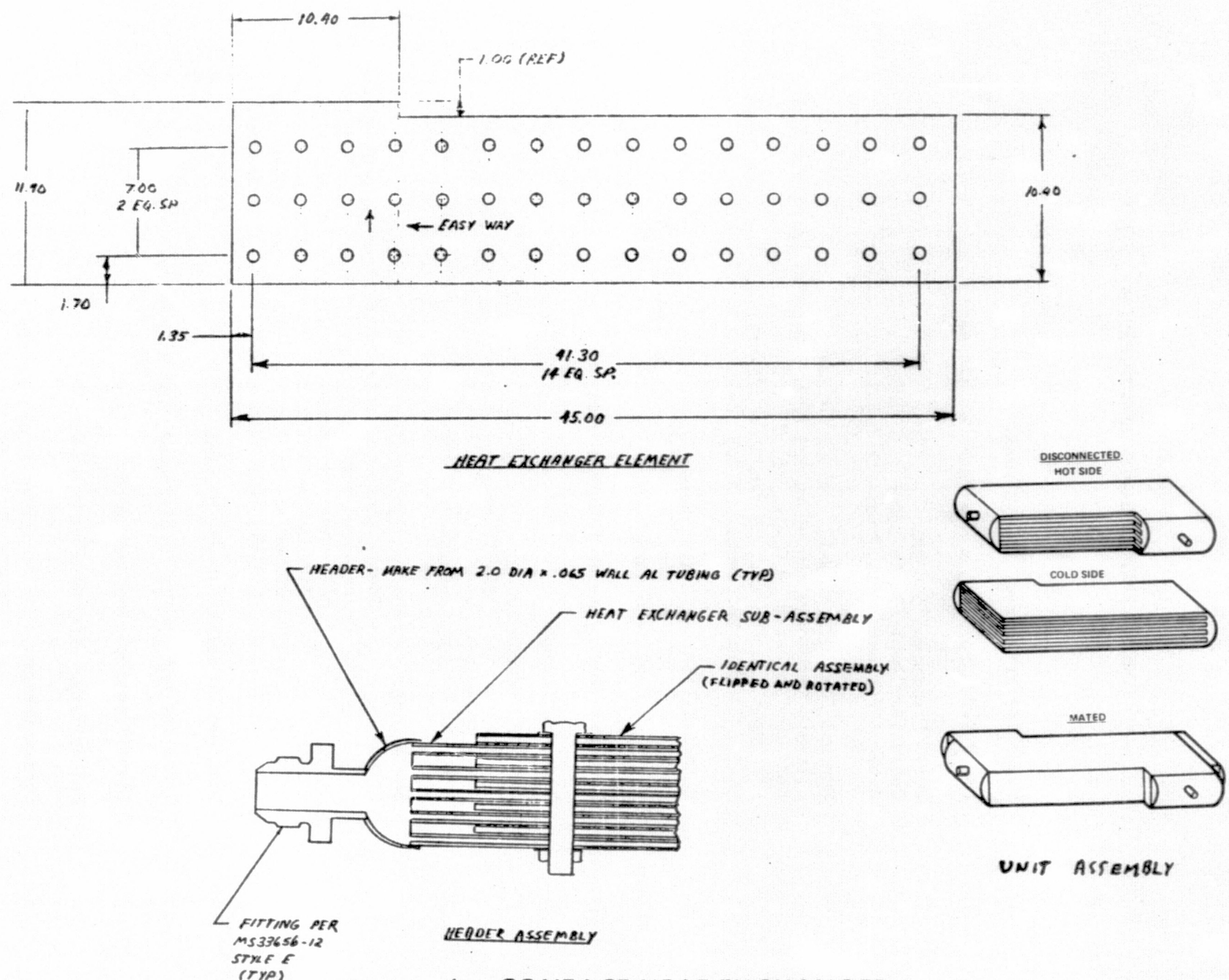

FIGURE 4 CONTACT HEAT EXCHANGER

TABLE I

CONTACT HEAT EXCHANGER SPECIFICATION

1. Operating Pressure: 250 psi max.

2. Proof Pressure: 1.5 x Max Oper = 375 psi

3. Burst Pressure: 500 psi
   Operating Temp: 30 to 150°F
   Ambient Pressure: 14.7 psia to 0 psia

4. Surface Finish: 40 μ-in. rms or better
   (plate thickness can be varied from 0.040" to improve finish)
   Surface Flatness: 0.01-in./ft. T.I.R.

5. Fin Type: 20 fin/in.; rectangular off-set; 1/8-in. flow length
   Fin Thickness: 0.005 inch.

6. Material: No. 11 Braze Sheet, 3003 Al Fin, 6061-T6 Header — or Equivalent

7. Process: Dip braze or fluxless braze

8. Flow Conditions:

   Fluid: Freon 12
   Max Flowrate: 3300 lb/hr
   Max Allowable ΔP = 5 psi
   Min h = 200 B/hr-ft^2-°F

   Deviations:
   - Use vacuum fluxless or dip brazing
   - Change plate thickness as desired to produce the required surface finish
   - Will consider changes to ΔP & h_i if specified - Give us new width and length. (Please be aware of contact conductance heat transfer between plates; this increases size of HX considerably)

9. Leakage: 0 cc/min at 250 psi and 150°F

10. Vibration and Shock: satisfactory for laboratory use

11. Identification: not required

12. Reliability: 20,000 hr MTBF

13. Design Changes: no change which may affect form, fit, function, materials, or processes without Vought Systems Division approval

14. Interchangeability: not required

15. Storage Life: 5 years

16. Quality Assurance:

   - Vought may have representative present during fab.
   - Follow good practice

17. Acceptance Tests

   - Vought may conduct tests, not to exceed requirements of this spec.
   - We are not requiring that they quarantee design, so we can only check ΔP & h_i

18. Proof Pressure Test: unit to withstand proof pressure for 15 minutes, no deformation or permanent set

19. Cleanliness: good practice in fabrication area

20. Corrosion: if salt pot brazing is used - all salt will be flushed from system

21. Similarity Test Data: If supplier intends to claim similarity the supplier shall supply one copy of test report applicable to tests for which similarities claimed.

22. Pressure Cycling: not required

23: Ref. Drawing 221-60053 for configuration

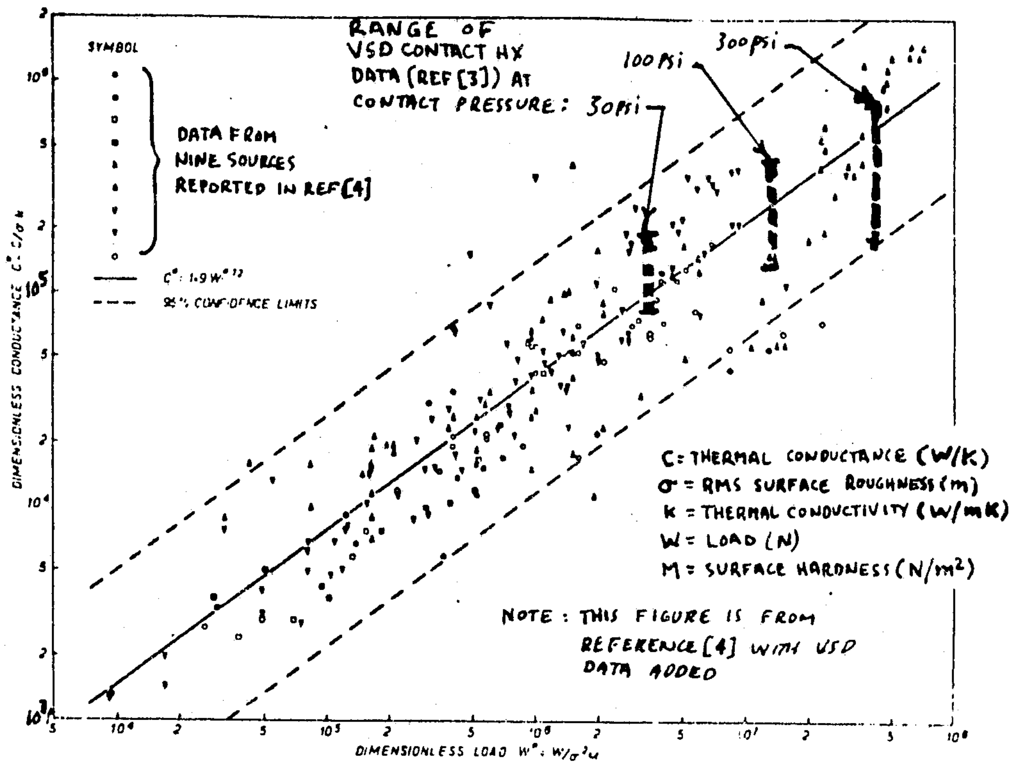

FIGURE 5 VSD CONTACT HEAT EXCHANGER DATA CORRELATION CONSIDERING SURFACE ROUGHNESS

facility can accommodate.

Evaluation of the contact heat exchanger quotes is underway, using the following evaluation criteria:

CRITERIA	WEIGHTING FACTOR
(1) Price	25%
(2) Proposed Approaches	25%
(3) Technical Capability	12-1/2%
(4) Vendors experience with similar hardware	12-1/2%
(5) Schedule	12-1/2%
(6) Vendor's reputation for meeting committments	12-1/2%
	100%

3.0 CURRENT PROBLEM AREAS

There are two major problem areas at this time:

(1) A decision is needed by 1 February 1975 from NASA-JSC on the inverter to be used in the flight prototype SHRM test, and

(2) There may be a problem in obtaining the contact heat exchanger in time for the flight prototype test. Schedule considerations could force the choice in the vendor.

4.0 WORK PLANNED FOR THE NEXT REPORTING PERIOD

The work planned for the next reporting period includes:

(1) Fabrication and delivery of the four SHRM radiator panels to NASA-JSC

(2) Fabrication and delivery of the eight cryogenic fluid swivels to NASA-JSC

(3) Selection of a contact heat exchanger vendor

(4) Refinement of the SHRM control system

(5) Completion of hardware definition for the flight prototype SHRM

(6) Initiation of definition of the VSD inputs to the SHRM test requirements document

and (7) Development of the SHRM test timeline

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5.0 REFERENCES

[1] Contract NAS9-14408, "Development of a Self-Contained Heat Rejection Module" NASA-JSC, dated 12 December 1974.

[2] VSD Report No. T211-RP004, "Self-Contained Heat Rejection System Progress Report", dated 7 December 1973.

[3] Leach, J. W., "Joint Conductance Element Test for SHRM contact Heat Exchanger", VSD Report T211-RP-009, dated 15 August 1974.

[4] Thomas, T. R. and S. D. Probert, "Correlations for Thermal Contact Conductance in Vacuo", Trans of ASME:J. of Heat Transfer, August 1972, pp 276-281.

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